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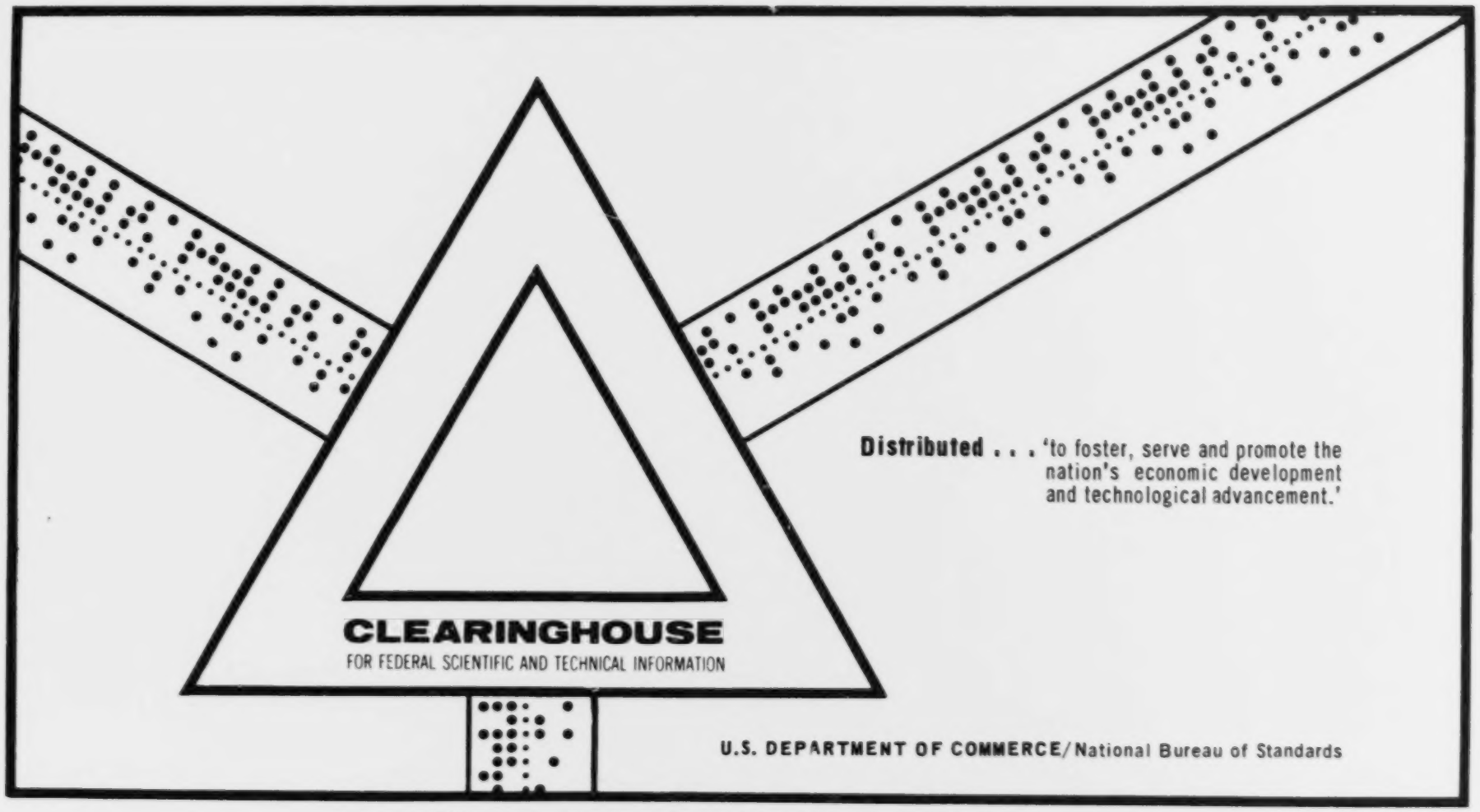
RESIDUAL FUEL OIL-WATER EMULSIONS

R. E. Barrett, et al

Battelle Memorial Institute
Columbus, Ohio

12 January 1970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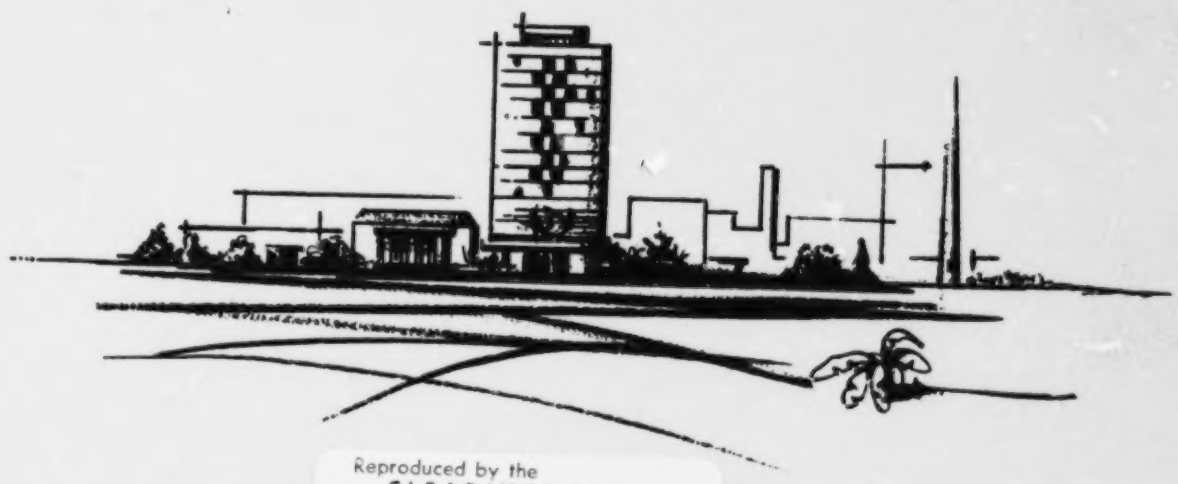

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SUMMARY REPORT

on

RESIDUAL FUEL OIL-WATER EMULSIONS

Contract No. PH 86-68-84
Task Order No. 16

to

NATIONAL AIR POLLUTION CONTROL ADMINISTRATION
Division of Process Control Engineering

January 12, 1970

by

R. E. Barrett, J. W. Moody, H. R. Hazard,
A. A. Putnam, and D. W. Locklin

BATTELLE MEMORIAL INSTITUTE
Columbus Laboratories
505 King Avenue
Columbus, Ohio 43201

A B S T R A C T

This task report describes the development of techniques for the preparation of emulsions of No. 6 residual fuel oil and water. It also includes the preliminary design of a combustion test rig for studying the combustion of heavy oils and emulsions and pollutant emission produced by burning these fuels.

Stable emulsions containing up to 30 percent water, by weight, were prepared from 9 samples of residual fuel oils. No emulsifying agents were required as the oils apparently contained natural emulsifiers. Emulsions containing basic additives were also prepared. The emulsions did not appear to break during atomization.

The preliminary design of the combustion test rig showed that a rig could be constructed to operate with either combustion intensity or pressure drop comparable to that of a full-scale unit. This can be accomplished by varying the firing rate from 1.5 gph for constant combustion intensity firing to about 8.0 gph for constant pressure drop firing. A variable swirl-type burner was designed to provide the flexibility needed for this range of combustion conditions.

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January 12, 1970

National Air Pollution Control Administration
3914 Virginia Avenue
Cincinnati, Ohio 45226

Attention Mr. John H. Wasser, Project Officer
Division of Process Control Engineering

Contract No. PH 86-68-84
Task Order No. 16

Gentlemen:

We have completed our assignment under the subject task order and hereby enclose 20 copies of our summary report, "Residual Fuel Oil-Water Emulsions".

The report outlines the procedure developed for preparation of residual fuel oil-water emulsions and contains data on certain important properties (particularly viscosity) of the emulsions. Modeling criteria for simulating industrial combustion systems on a laboratory scale are discussed, and a preliminary design of a residual fuel-oil combustion test rig is presented.

If you have any questions, we would be pleased to discuss any of the points in greater detail.

Sincerely,

D. W. Locklin
Associate Chief
Thermal Systems Division

DWL:jc

Enc. (20)

cc: Mr. R. P. Hangebrauck

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SUMMARY REPORT

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INTRODUCTION

The Division of Process Control Engineering of the National Air Pollution Control Administration is presently evaluating methods of reducing air-pollutant emissions from combustion devices including oil burners for space heating.

For the domestic oil burner part of this program, NAPCA is conducting tests in a furnace which utilizes a 1-gph high-pressure atomizing burner firing commercial No. 2 fuel oil[1,2]. These tests will determine the extent to which air pollutants can be reduced by using various fuel additives in domestic-size oil-heating units. Battelle provided support for the NAPCA program in an earlier task[3] that resulted in development of preparation techniques for water-in-distillate fuel-oil emulsions and preliminary firing of these emulsions. Subsequent studies by NAPCA burning emulsified distillate fuel oil produced evidence that some pollutant emissions (specifically NO_x) were reduced.

In view of these results, NAPCA extended its interest in emulsified fuels to include residual fuel oil. Because of the greater difficulty in obtaining good atomization of heavy fuel oils, secondary atomization due to emulsification may produce greater benefits (such as reduction of hycrocarbon, carbon monoxide, and/or particulate emissions) when used with these fuels.

TASK OBJECTIVE

Battelle's assignment under this task order was to develop techniques for preparing emulsions of water in typical residual fuel oils, to describe the

relative merits and problems encountered in the practical application of residual fuel oil emulsions, and to prepare a preliminary design and cost estimate for construction of a laboratory-scale residual-oil-fired combustion test rig.

SUMMARY AND CONCLUSIONS

Relatively stable water-in-residual fuel oil emulsions have been prepared from 9 residual oil samples. Preparation of these emulsions did not require the addition of an emulsifying agent because the oils apparently contained sufficient natural emulsifiers. Several chemically basic materials, which might react with sulfur or sulfur compounds in a furnace environment, were also incorporated into the emulsions.

Experimental results indicate that the emulsions remain stable when stored at temperatures of 6 F for 2 hours and 160 F for 3 days. Results of a brief atomization trial of a single emulsion suggested that the emulsion was not broken during atomization.

Viscosity of the residual fuel oils and the emulsions was measured at temperatures of 100, 122, and 160 F and at three shear rates. Each of the oils and most emulsions exhibited near-Neutonian properties. The emulsions were consistently more viscous than the residual oils from which they were made. Approximately 20 F additional preheat will be necessary when firing emulsions to obtain viscosities equal to the viscosities at which residual oils are typically atomized.

No inherent property of the emulsions would rule out their use as fuels. The potential benefits of improved combustion through better atomization, possibly resulting in lower pollutant emissions, seems to warrant further investigation of the use of emulsified residual fuel oils. Specifically, the program should be extended to combustion trials and emission measurements.

The preliminary design of a combustion test rig in which to conduct this investigation is described in this report. It appears that a practical laboratory rig can be constructed to fire about 1.5 gph, but that a special burner will be required to provide the needed flexibility. The estimated cost to construct and instrument this rig is about $48,000. This cost estimate does not include costs associated with further burner development or gas analysis and emission measurement.

BACKGROUND

NAPCA became interested in evaluating emulsified oil-water fuels as a result of improved combustion (evidenced by shorter combustion times) reported by other investigators. The concept of firing fuel oil-water emulsions to obtain secondary atomization and, therefore, improve combustion was advanced by Ivanov and others[4] in 1957, and further discussed by Ivanov and Nefedov[5] in 1962. In the desired type of oil-water emulsion, each fuel oil droplet contained one or more small droplets of water. As the emulsion was sprayed into a hot chamber, they found that the water within the fuel droplet vaporized before the fuel was consumed. The pressure generated within the fuel droplet by the vaporizing water was sufficient to rupture the fuel droplet in a miniature explosion. This shattering or secondary atomization of the fuel droplet produces further reduction in the effective droplet size and exposes a greater surface area of fuel for vaporization, mixing, and burning, thus providing more rapid and improved combustion.

In a previous program[3], a method for preparing stable water-in-distillate fuel oil emulsions was developed and these emulsions were fired in a burner similar to that used by NAPCA. In subsequent work at the NAPCA Division of Process Control Engineering, measurements showed evidence that emulsification of water in No. 2 fuel oil could reduce NO_x emissions. The work described here is an extension of that previous program to include the study of emulsions of water-in-residual fuel oils.

Most of the literature on water-petroleum emulsions deals with methods of breaking oil-field emulsions or with the commercially important oil-in-water

bituminous emulsions used in road work. However, emulsified fuels have been the subject of a number of patents and papers over the years. Of prime pertinence to this program is the aforementioned work of Ivanov et al[4-7] who suggested the concept of improving combustion processes by emulsifying water in fuel oils. Also of interest is the work of Cornet and Nero[8] and Maillard[9] who demonstrated that emulsification of water in diesel fuels increased the efficiency of an engine under certain conditions and reduced the amount of smoke in the exhaust and the amount of carbon deposit in the engine. The preparation of water-in-fuel oil emulsion has been described in a number of patents[10] and, recently, water-in-oil emulsions have been developed as fire-resistant hydraulic fluids[11-12]. Oil-in-water emulsions are currently being investigated for use as a fire resistant fuels for aircraft[13-26].

CHARACTERISTICS OF RESIDUAL FUEL OILS

Properties of residual fuel oils vary widely depending on crude source, refinery processing, and blending. Specifications for residual oils are established in ASTM Standard D-396 for grades as follows:

Grade	Viscosity Range SSU @ 100 F	Preheating
No. 5 (light)	150-300	requires little or no preheating for firing
No. 5 (heavy)	350-750	generally requires preheating for firing and often for handling
No. 6	900-9000	requires preheating for both firing and handling

Fuels within these grades require different equipment and different levels of preheating to reduce the viscosity necessary for handling and firing.

Table 1 shows the complete ASTM specifications for all grades. It should be noted that the range of properties permitted by the ASTM standard is much greater for No. 6 grade than the No. 5 grades. (No. 4 oil is sometimes considered as a residual fuel oil, but is not included in the scope of this study.)

An example of the variation in properties that are introduced by crude source is shown in Table 2 from data presented by Siegmund[27]. Besides the differences in listed properties of the crudes, there are also variations in basic crude composition; for example, the Tia Juano medium crude is a low wax crude and the Libyan crude is a paraffin or high-wax crude. Processing variations, such as the several desulfurization processes, can also introduce considerable variation into properties of residual fuel oils.

Important properties of finished fuels can be controlled by blending of residual from different crudes and/or by blending with distillate stocks to reduce viscosity, pour point, or sulfur. The varying degree of distillate blending adds to the wide variations in properties of No. 6 oil observed in the marketplace.

Fuel properties are likely to change somewhat in the next few years. Air pollution regulations are already limiting suflur content of fuel oils in many areas and this trend will most certainly increase. For fuels used in New York City, sulfur content is now limited to 1.0 percent and in a few years the limit is scheduled to drop to 0.3 to 0.4 percent. To meet such requirements, refiners can control sulfur content to some extent by a combination of crude selection, processing, and blending.

TABLE 1. ASTM D-396 SPECIFICATIONS FOR FUEL OILS

Grade of Fuel Oil [a]		Flash Point, deg F (deg C)	Pour Point, deg F (deg C)	Water and Sediment, per cent by volume	Carbon Residue on 10 per cent Bottoms, per cent	Ash, per cent by weight	Distillation Temperatures, deg F (deg C)			Saybolt Viscosity, sec				Kinematic Viscosity, centistokes				Gravity, deg API	Copper Strip Corrosion	Sulfur per cent
							10 per cent Point	90 per cent Point		Universal at 100 F (38 C)		Furol at 122 F (50 C)		At 100 F (38 C)		At 122 F (50 C)				
		Min	Max	Max	Max	Max	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Max
No. 1	A distillate oil intended for vaporizing pot-type burners and other burners requiring this grade of fuel	100 or legal (38)	0[d]	trace	0.15	...	420 (215)	...	550 (288)	...	...	...	...	1.4	2.2	..	...	35	No. 3	0.5 or legal
No. 2	A distillate oil for general purpose domestic heating for use in burners not requiring No. 1 fuel oil	100 or legal (38)	20[d] (−7)	0.05	0.35	...	[e]	540[d] (282)	640 (338)	(32.6)[f]	(37.93)	...	...	2.0[d]	3.6	..	...	30		0.5[b] or legal
No. 4	Preheating not usually required for handling or burning	130 or legal (55)	20 (−7)	0.50	...	0.10	...	...	...	45	125	...	...	(5.8)	(26.4)	...	...		..	[c]
No. 5 (Light)	Preheating may be required depending on climate and equipment	130 or legal (55)		1.00	...	0.10	...	...	...	150	300	...	...	(32)	(65)	...	...		...	[c]
No. 5 (Heavy)	Preheating may be required for burning and, in cold climates, may be required for handling	130 or legal (55)		1.00	...	0.10	...	...	...	350	750	(23)	(40)	(75)	(162)	(42)	(81)	..	...	[c]
No. 6	Preheating required for burning and handling	150 (65)	..	2.00[g]	...	...	...	...	...	(900)	(9000)	45	300	...	...	(92)	(638)		...	[c]

[a] It is the intent of these classifications that failure to meet any requirement of a given grade does not automatically place an oil in the next lower grade unless in fact it meets all requirements of the lower grade.

[b] Outside USA the sulfur limit for No. 2 shall be 1.0 percent.

[c] Legal requirements to be met.

[d] Lower or higher pour points may be specified whenever required by conditions of storage or use. When pour point less than 0 F is specified, the minimum viscosity shall be 1.8 cSt (32.0 sec, Saybolt Universal) and the minimum 90 per cent point shall be waived.

[e] The 10 per cent distillation temperature point may be specified at 440 F (226 C) maximum for use in other than atomizing burners.

[f] Viscosity values in parentheses are for information only and not necessarily limiting.

[g] The amount of water by distillation plus the sediment by extraction shall not exceed 2.00 per cent. The amount of sediment by extraction shall not exceed 0.50 per cent. A deduction in quantity shall be made for all water and sediment in excess of 1.0 per cent.

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TABLE 2. CHARACTERISTICS OF MIDDLE EAST AND VENEZUELAN CRUDES AND FUEL OILS[27]

	Light Arabian		Kuwait		Tia Juana Med		Libyan	
	Crude	Fuel Oil	Crude	Fuel Oil	Crude	Fuel Oil	Crude	Fuel Oil
API Gravity	34.7	15.5	31.4	15.5	26.5	16.3	39.2	22.2
Sulfur, Wt %	1.7	3.0	2.5	4.0	1.5	2.0	0.2	0.4
Pour Point, °F	-15	+55	-20	+55	-40	+20	+40	+105
Viscosity SSU @ 100°F	44	--	56	--	116	--	40.5	--
SSF @ 122°F	--	175	—	175	--	170	--	65
Vanadium PPM	13	37	31	61	156	240	3	10
Nickel PPM	4	11	7	14	20	31	—	—

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TABLE 3. SUMMARY OF GRADE 5 (light) FUELS(28)

Geographic distribution of burner fuel oils 1/ / Districts within region / Additional districts 2/ / Number of fuels				Eastern region A,B,C / D,E,F,G,J / 7			Southern region D / B,G,J / 1	Central region E,F,G / B,C,D,I,J,K,L / 10			Rocky Mountain region H,I,J,K / B,D,E,F,G,L / 4			Western region L,M,N,O,P / E,F,G,I,K / 1
Test	ASTM D396 requirements Method	Min.	Max.	Minimum	Average	Maximum	(3)	Minimum	Average	Maximum	Minimum	Average	Maximum	(3)
Gravity — ° API	D287	-	-	15.4	18.8	20.9	16.4	9.2	16.7	20.7	5.5	12.4	16.4	13.8
Flash point, Pensky-Martens closed tester — ° F.	D93	130	-	170	-	208	208	180	-	230	190	-	230	230
Viscosity														
Kinematic at 100° F. — centistokes	D445	32	65	33.8	47.3	69	37.5	33.8	48.0	69	37.5	54.0	68.7	60
Furol at 122° F. — seconds	D88	-	-	-	-	-	-	18	(4)	21		3/17.3		-
Pour point — ° F.	D97	-	-	-40	-	35	-5	-15	-	35	-10	-	75	-10
Sulfur content — percent	D129	-	-	0.68	1.01	1.4	1.36	0.64	1.33	3.4	1.36	2.17	3.4	3.4
Ramsbottom carbon residue on 100 percent sample — do.	D524	-	-	3.3	7.1	10.5	6.3	3.1	6.0	9.2	6.0	7.3	9.3	6.0
Ash — do.	D482	-	0.10	0.006	0.042	0.18	0.025	0.006	0.017	0.025	0.0095	0.016	0.025	0.02
Water and sediment — vol. percent	D1796	-	1.00	0.05	0.1	0.3	0.1	0.0	0.1	0.15	0.0	0.0	0.1	0.0

TABLE 4. SUMMARY OF GRADE 5 (heavy) FUELS(28)

Geographic distribution of burner fuel oils 1/ / Districts within region / Additional districts 2/ / Number of fuels				Eastern region A,B,C / E,F,G / 4			Southern region D / G / 2			Central region E,F,G / C,D,J / 7			Rocky Mountain region H,I,J,K / L,M / 6			Western region L,M,N,O,P / H,K / 13		
Test	ASTM D396 requirements Method	Min.	Max.	Minimum	Average	Maximum	Minimum	Average	Maximum	Minimum	Average	Maximum	Minimum	Average	Maximum	Minimum	Average	Maximum
Gravity — ° API	D287	-	-	15.3	16.4	18.3	4.6	12.7	17.1	4.6	15.2	18.3	3.6	10.8	17.8	7.6	12.7	17.6
Flash point, Pensky-Martens closed tester — ° F.	D93	130	-	148	-	208	176	-	>200	168	-	>200	194	-	240	162	-	240
Viscosity,																		
Kinematic at 100° F. — centistokes	D445	75	162	78.8	107.9	138.2	69	75.5	81.9	69	100.1	138.2	71.8	92.9	117.0	71.8	110.9	154
Furol at 122° F. — seconds	D88	23	40	-	-	-	27	30.0	33	27	3/31.0	33	22.3	27.7	33.9	21	28.7	37.5
Pour point — ° F.	D97	-	-	-200	-	70	-30	-	85	-30	-	85	-15	-	95	-25	-	95
Sulfur content — wt. percent	D129	-	-	0.55	1.23	2.30	1.44	2.13	3.25	0.55	1.73	3.25	1.48	2.18	2.98	0.82	1.45	2.98
Ramsbottom carbon residue on 100 percent sample — percent	D524	-	-	4.8	6.7	10.0	4.0	7.7	12.0	4.0	6.5	9.0	4.7	7.9	9.6	3.8	7.2	10.8
Ash — percent	D482	-	0.10	0.008	0.012	0.015	0.04	0.017	0.03	0.008	0.015	0.03	<0.01	0.020	0.06	0.017	0.035	0.08
Water and sediment — vol. percent	D1796	-	1.00	0.05	0.2	0.3	.08	0.1	0.2	0.05	0.1	0.3	0.1	0.2	0.2	0.0	0.1	0.2

Data From Bureau of Mines Survey, 1968

TABLE 5. SUMMARY OF GRADE 6 FUELS (28)

Geographic distribution of burner fuel oils 1/				Eastern region A,B,C D,E,F,G,J 35			Southern region D A,B,C,D,E,G,J 10			Central region E,F,G A,B,C,D,H,I,J,K,L 32			Rocky Mountain region H,I,J,K B,D,E,F,G,L,M,N 18			Western region L,M,N,O,P E,F,G,H,I,K 21		
Districts within region																		
Additional districts 2/																		
Number of fuels																		
Test	ASTM D396 requirements Method	Min.	Max.	Minimum	Average	Maximum	Minimum	Average	Maximum	Minimum	Average	Maximum	Minimum	Average	Maximum	Minimum	Average	Maximum
Gravity, ° API	D287	-	-	7.4	13.1	22.5	3.9	10.4	14.6	-4.7	10.1	18.1	-1.5	8.0	15.5	2.9	7.7	11.5
Flash point, Pensky-Martens closed tester, ° F.	D93	130	-	165	-	360	170	-	234	165	-	335	198	-	340	170	-	335
Viscosity																		
Furol at 122° F., seconds	D88	45	300	67	180.8	310	135	175.1	200	55	189.9	310	43.5	164.1	295	55	147.2	288
Pour point, ° F.	D97	-	-	10	-	100	10	-	90	10	-	90	10	-	100	15	-	100
Sulfur content, percent	D129	-	-	0.18	1.34	2.91	0.63	1.81	3.41	0.52	1.69	4.1	0.97	2.24	4.1	0.74	1.44	4.1
Ramsbottom carbon residue on 100 percent sample, do.	D524	-	-	2	11.4	18.3	5.3	14.1	18.3	8.4	10.6	18.3	6.3	11.0	16	5.9	12.3	17.4
Ash, do.	D482	-	-	0.005	0.048	0.185	0.01	0.046	0.183	0.008	0.030	0.10	<0.01	0.050	0.12	0.02	0.049	0.12
Water by distillation, vol. percent	D95	-	-	0.0	0.08	0.35	0.0	0.07	0.2	0.0	0.07	.25	0.0	0.05	0.2	0.0	0.05	0.2
Sediment by extraction, wt. percent	D473	-	0.50	.0	.07	0.4	0.02	.06	.2	.0	.08	0.4	.0	.12	.5	0.02	.11	.5

Data From Bureau of Mines Survey, 1968

Bureau of Mines Survey of
Burner Fuel Oils, 1968

The most comprehensive information on properties of fuels sold in the U.S. is provided by the Bureau of Mines annual survey of burner fuel oils[28]. This provides data on samples as marketed in different geographic regions.

Tables 3, 4, and 5 summarize data on No. 5 and No. 6 oils by marketing regions, showing minimum, arithmetic average, and maximum values of significant properties[28]. These cover 15 samples of No. 5 (light), 24 samples of No. 5 (heavy), and 81 samples of No. 6 oil. It is noted that there is a significant variation in average values of some properties from region to region and, also, that there is a wide range in the property values within any region.

The data are summarized on a national basis in Tables 6, 7, and 8. The median value of properties is defined as that value where one half the samples were above and one half were below for the property of interest.

Figure 1 shows the effect of temperature on viscosity for the ranges of No. 5 and No. 6 oils reported in the survey. Median values are also shown.

TABLE 6. NATIONAL SUMMARY OF PROPERTIES FOR NO. 5 (LIGHT) BURNER FUEL OIL (1968)

Fuel	Minimum	Median	Maximum
Gravity, °API	5.5	17.9	20.9
Flash point, F	170	200	230
Viscosity, kinematic at 100 F, cs	34	47	69
Pour point, F	-40	0	75
Sulfur content, wt. percent	0.64	1.25	3.4
Carbon residue, percent	3.1	6.4	10.5
Ash, wt. percent	.006	.016	.18
Water and sediment, vol. percent	.0	.1	.3

TABLE 7. NATIONAL SUMMARY OF PROPERTIES FOR NO. 5 (HEAVY) BURNER FUEL OIL (1968)

Fuel	Minimum	Median	Maximum
Gravity, °API	4.6	13.8	18.3
Flash point, F	162	198	240
Viscosity, kinematic at 100 F	69	105	154
Viscosity, Furol at 122 F,sec	21	27	37.5
Pour point, F	-30	0	95
Sulfur content, wt. percent	.55	1.52	3.25
Carbon residue, percent	3.8	7.5	10.8
Ash, wt. percent	.008	0.02	.08
Water and sediment, vol. percent	.05	0.1	.3

TABLE 8. NATIONAL SUMMARY OF PROPERTIES FOR NO. 6 BURNER FUEL OIL (1968)

Fuel	Minimum	Median	Maximum
Gravity, °API	-4.7	10.5	22.5
Flash point, F	165	209	395
Viscosity, Furol at 122 F,sec	43.5	178	310
Pour point, F	10	35	100
Sulfur content, wt. percent	.18	1.33	3.25
Carbon residue, percent	.6	11.6	18
Ash, wt. percent	.003	.03	.35
Water by distillation, vol.percent	.0	.05	.35
Sediment by extraction, wt.percent	.0	.05	.5

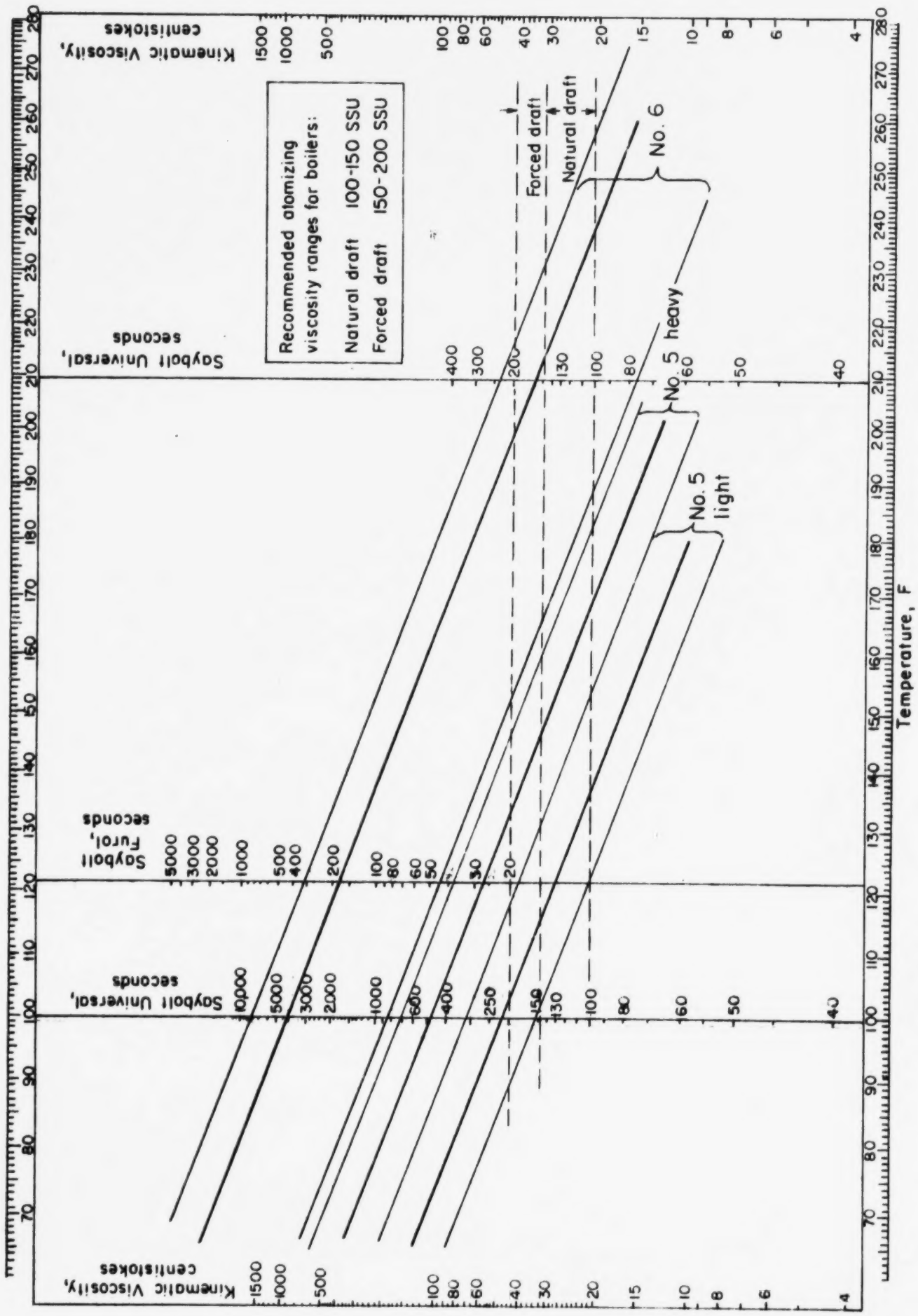

FIGURE 1. VISCOSITY-TEMPERATURE RELATION FOR RESIDUAL FUEL OILS IN 1968 BUREAU OF MINES SURVEY

In an attempt to define "typical" oils within the No. 6 grade, several plots were made of important properties for fuels reported in the Bureau of Mines Survey. These plots included:

sulfur content vs viscosity (Figure 2)
sulfur content vs gravity (Figure 3)
gravity vs viscosity (Figure 4).

The only plot which may be interpreted as showing groupings of residual oils is sulfur content vs viscosity. Figure 2 shows this plot with the major oil grouping shown by dotted lines. Four groups of oils may be identified as follows:

- low viscosity, low sulfur (9 oils)
- medium viscosity, low to medium sulfur (34 oils)
- medium viscosity, high sulfur (9 oils)
- high viscosity, low to medium sulfur (10 oils).

Table 9 gives the median viscosity and sulfur content for each of these four groups of residual oils.

TABLE 9. MEDIAN PROPERTIES OF FOUR GROUPS OF RESIDUAL OILS

Fuel	Low visc., low suflur	Med. visc., low to med. sulfur	Medium visc., high sulfur	High visc., low to med. sulfur
Viscosity, Saybolt Furol at 122 F, sec	106	175	180	261
Sulfur content, wt. percent	0.81	1.27	2.64	0.90

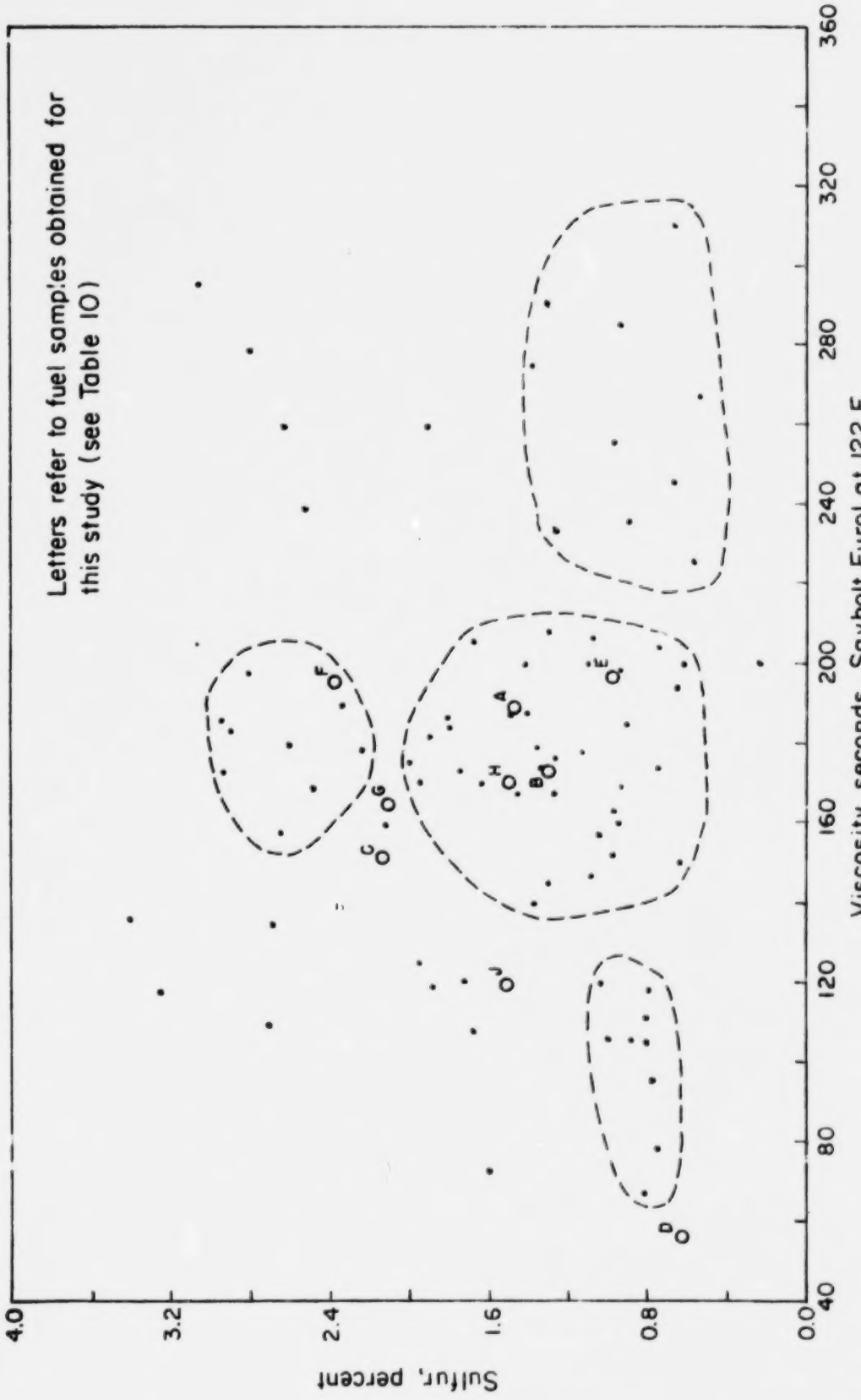

FIGURE 2. SULFUR CONTENT VERSUS VISCOSITY FOR NO. 6 RESIDUAL FUEL OILS

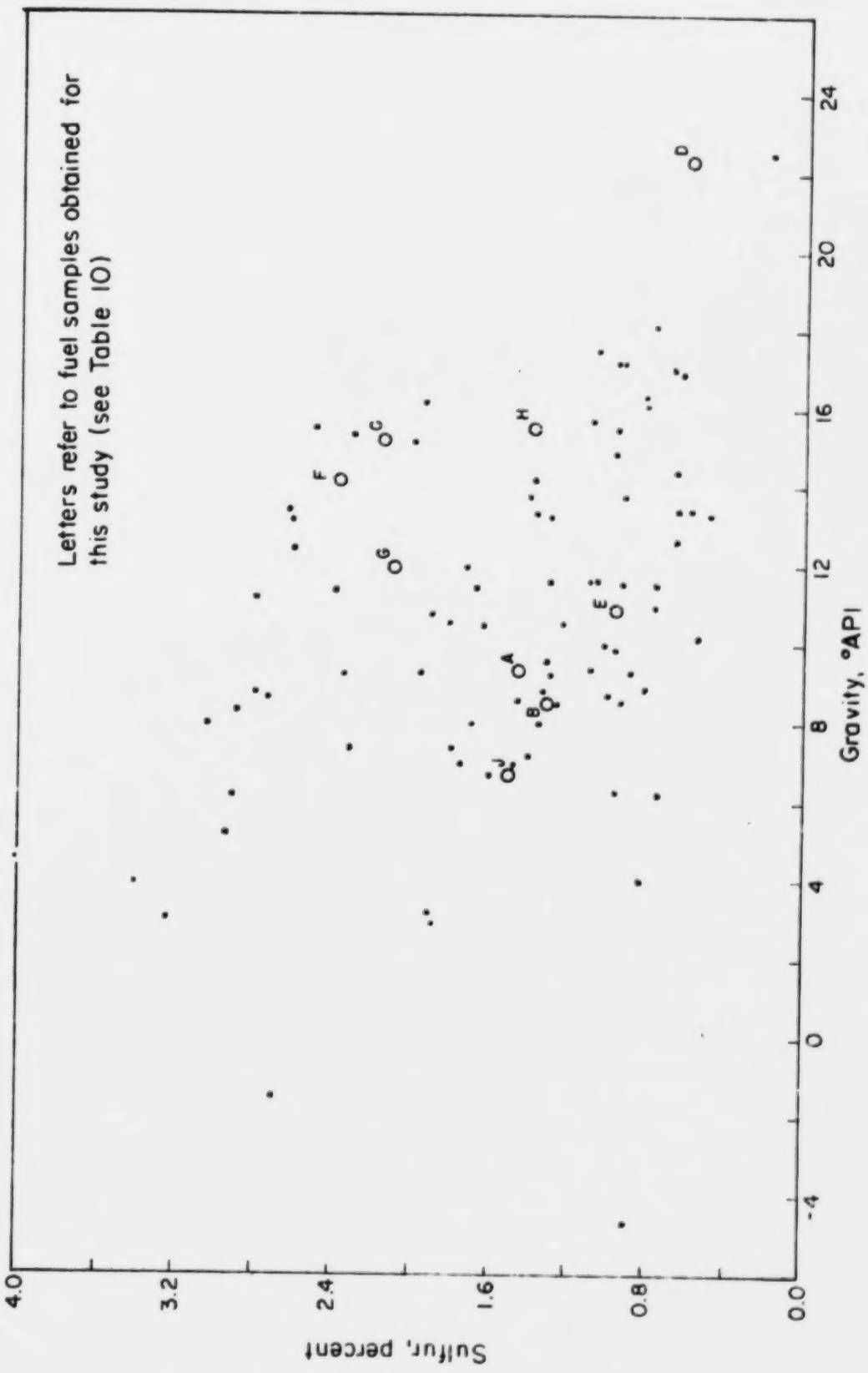

FIGURE 3. SULFUR CONTENT VERSUS GRAVITY FOR NO. 6 RESIDUAL FUEL OILS

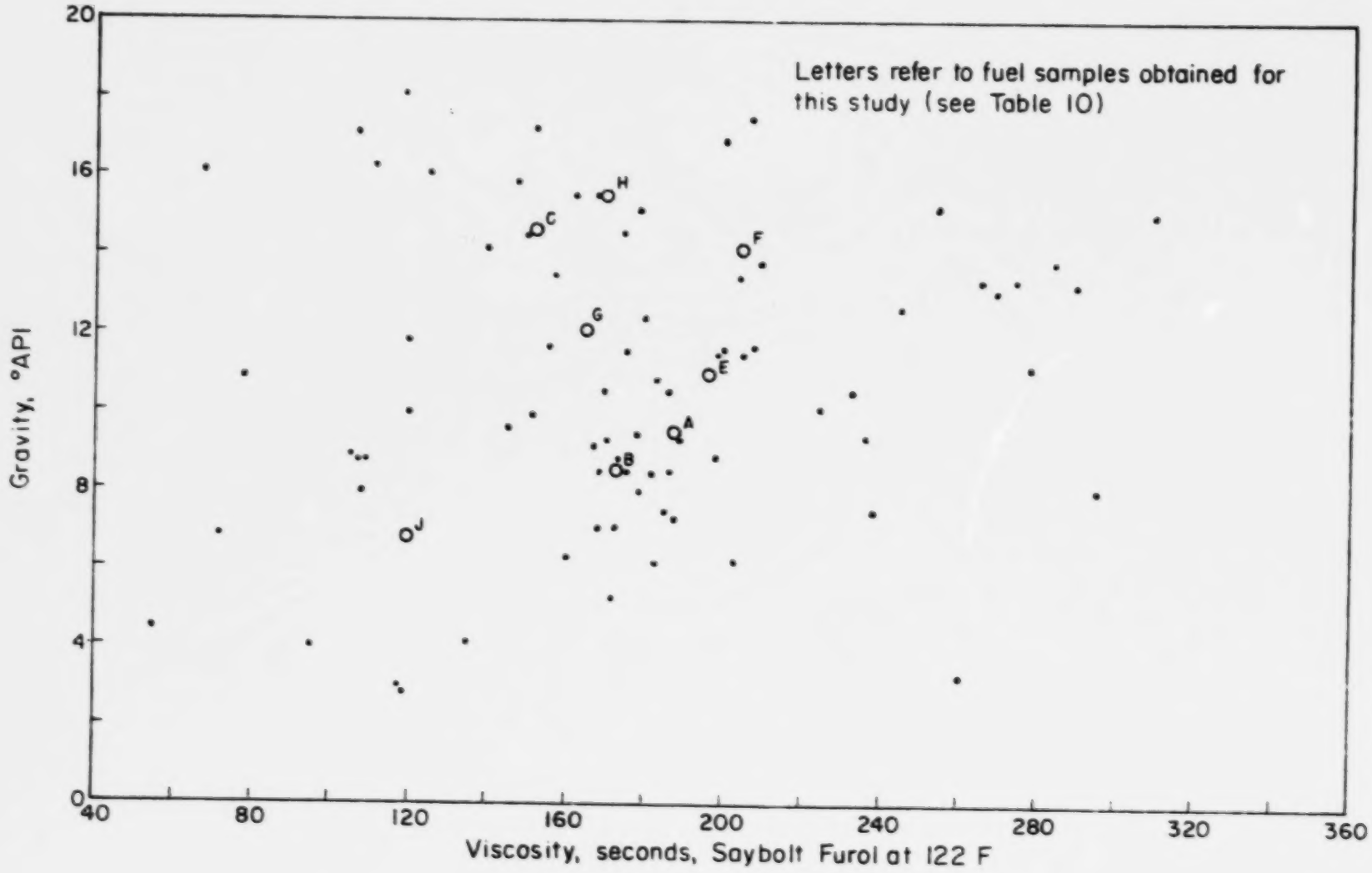

FIGURE 4. GRAVITY VERSUS VISCOSITY FOR NO. 6 RESIDUAL FUEL OILS

Residual Fuel Samples Obtained For This Study

As part of this study, five refiners were contacted and asked to supply sample quantities (5 gallons) of typical No. 6 residual fuel oils for the emulsification trials. Nine samples were supplied by the refiners.

Table 10 lists the properties of the nine oils as supplied by the refiners. Figures 2, 3, and 4 show how these oils compare to all No. 6 residual fuel oils marketed during 1968. It can be seen from these plots that at least seven of the oils obtained for this program were generally characteristic of "average" No. 6 fuel oils.

EXPERIMENTAL PROCEDURES AND RESULTS

The experimental portion of this study consisted of preparing water-in-oil emulsions, determining basic properties of the emulsions, and conducting "cold" atomization trials (without flame) to determine if the emulsions break during atomization.

Preparation of Water-in-Oil Emulsions

In the previous work on water-in-distillate fuel oil emulsions[3] it was found that a surfactant was required to produce stable emulsions. Early in this study it became apparent that very stable water-in-residual fuel oil emulsions could be prepared by simply mixing water and No. 6 grade fuel oils with sufficient agitation. This is not surprising when it is recalled that a large percentage of crude oils are obtained from the wells as water-in-oil emulsions.

Crude oil contains a number of constituents which tend to promote the formation of emulsions when the oil and water are mixed. Among the most common substances are naphthenic acids and their salts, asphalt particles,

TABLE 10. PROPERTIES OF RESIDUAL-OIL SAMPLES USED IN EMULSION EXPERIMENTS[a]

Oil Sample	A	B	C	D	E	F	G	H	J
Crude source	Mid-Cont.	W.Coast	___	Africa	Mid-Cont.	Venezuela	Gulf Coast	Venezuela	W.Coast
Properties									
Gravity, °API	9.3	8.3	14.6	22.3	10.9	14.1	12.0	15.5	6.8[b]
Flash point, F	315			260	225	206	155		265
Viscosity, Saybolt sec.Furol at 122 F	188[c]	173	151.	56.3	196.3	195.4	165	170[b]	120[b]
Pour point, F	30-40		15	95	25	20	35		5
Sulfur content, wt.pct.	1.45	1.3	2.14	0.59	0.96	2.37	2.1	1.5	1.5
Carbon residue, wt.pct.				6.65	8.9	9.6			12.6[b]
Ash, wt.pct.	0.22	0.09		0.004	0.05	0.05	0.02		0.06
Water & sediment, vol.pct.	0.04	0.3		0.10	0.20	0.10	0.2		
Vanadium, ppm	117.5		208						
Sodium, ppm	18.0		41					236	79
Lead, ppm	<1.0								
Manganese, ppm	<1.0								
Magnesium, ppm	6.2		12						
Calcium, ppm	34.4		21						
Nickel, ppm	26.9		81						
Iron, ppm			<6					86	78

(a) Properties supplied by refiners

(b) Converted from other units

(c) From Battelle data

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microscopic paraffin flakes, certain heavy hydrocarbons and clay or earthy matter. Many of these constituents, particularly the naphthenic acids and heavy hydrocarbons, are also found in residual fuel oils. The presence of these materials accounts for the ease with which water-in-oil emulsions can be formed with these oils.

However, the composition of crude oils and, hence of residual fuel oils vary considerably according to their geographic source. The method of refining also affects the composition of residual fuel oils. It is not to be expected, then, that the same type and concentration of naturally occurring emulsifiers will be found in all residual fuel oils. Therefore, it was important in this work to examine fuel oils from a number of sources to determine how common is the emulsification behavior of residual fuel oils.

It was found that each of the No. 6 grade fuel oils formed water-in-oil emulsions without the use of added emulsifying agents. The concentration range of stable emulsions was found to depend strongly on the method and temperature of mixing. Figure 5 illustrates the effects of these variables on the stable concentration range of emulsions formed with Oil A. Unstable emulsions immediately broke into two layers: (1) a relatively dilute oil-in-water emulsion which, in most cases, settled to the bottom of the container, and (2) a water-in-oil emulsion.

In Figure 5, the percent of the total volume occupied by the water-in-oil layer is plotted as a function of the concentration (weight percent) of water in the total system. The only stable emulsions are those which remained 100 percent water-in-oil emulsions. Curve 1 of the figure was generated by adding all the water in one step to the oil and mixing in a Waring Blendor. The mixture was stirred, at room temperature for 2 to 3 minutes. Under these conditions, stable emulsions containing up to about 35 weight percent water in the internal phase could be prepared. The stable concentration range extends to at least 50 weight percent if the water is added dropwise at room temperature

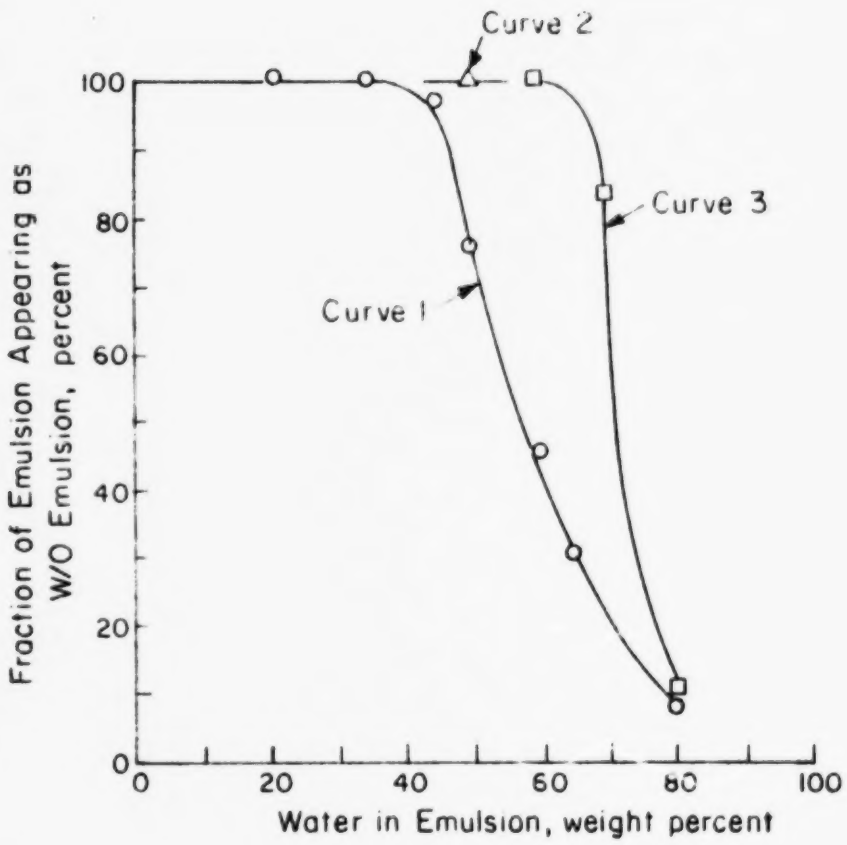

FIGURE 5. EFFECT OF MIXING VARIABLES ON STABLE CONCENTRATION RANGE OF OIL A EMULSIONS

- o Curve 1. Water added in one step, room temperature
- △ Curve 2. Water added dropwise, room temperature
- □ Curve 3. Water added in one step, oil 150-200 F, water boiling

as shown by the one point plotted for Curve 2. (Curve 2 is really an extension of Curve 1.) At this point the emulsion became so viscous that stirring on the Blendor became difficult. Emulsions containing up to about 60 weight percent water were prepared when the components were heated, as shown by Curve 3. In these experiments, boiling water was added, in one step, to Oil A which was pre-heated to 150 F to 200 F. The mixtures were then stirred 2 to 3 minutes on the Blendor.

The stable concentration range which might be obtained with all the oils is indicated by the data of Table 11. These data were obtained by adding the water in one step at room temperature (except where noted) to the oil on the Waring Blendor and mixing for 2 to 3 minutes. Although some difference in the behavior of the various oils is noted, the data indicate that emulsions containing up to about 30 weight percent water in the internal phase can be prepared easily with all the oils. The concentration range of stable emulsions could probably be extended in every case by modifying the mixing conditions. Sufficient oils were included in these experiments that it is felt that these conclusions can be extended to all common residual fuel oils.

TABLE 11. STABILITY CONCENTRATION RANGE OF WATER-IN-OIL EMULSIONS*

Water Weight Percent	Volume appearing as W/O Emulsion, Percent of total								
	Oil A	Oil B	Oil C	Oil D**	Oil E	Oil F	Oil G	Oil H	Oil J
20	100	100	100	100	100	100	100	100	100
40	97	99	93	100	100	99	92	90	100
60	45	60	53	100	50	54	50	67	81

* Emulsions prepared at room temperature (except for Oil D) by adding water in one step to oil on Waring Blendor and mixing 2 to 3 minutes.

** Emulsions with Oil D mixed at 150 F to 200 F.

Identification of Emulsion Type

Identifying the type of emulsion (water-in-oil or oil-in-water) obtained when the components were mixed was difficult. The dye tests which were so valuable with the water-distillate fuel oil studies[3] were useless in this work because of the dark color of the residual oils. Nor could definite conclusions be obtained from microscopic examination because of the rapid changes which occurred when the emulsions were spread on a microscope slide. In addition, the Waring Blendor whipped air into the emulsions and it was difficult to distinguish an air bubble from a water droplet in the internal phase.

Therefore, a number of other tests were applied to establish the emulsion type. These tests included:

(1) Color: Water-in-oil emulsions were black or dark brown while oil-in-water emulsions were a lighter brown.
(2) Phase Dilution: A drop of oil or water was floated on a few cc of the emulsion. If oil were the continuous phase, the oil drop would disperse but not the water drop. If water were the continuous phase, only the water drop would disperse. Since sometimes the water droplet would sink through the emulsion, great care had to be taken in interpreting this test.
(3) Filter Paper Wetting: A drop of emulsion was placed on a piece of filter paper. Oil-in-water emulsions spread rapidly, often leaving a dark spot of oil near the center of the area.

In combination, and with experience, these tests were sufficient to distinguish the emulsion type.

Incorporation of Basic Additives in Water-in-Residual Oil Emulsions

NAPCA suggested that basic additives might be included in the emulsions. These additives might react during combustion with sulfur from the fuel to reduce sulfur oxide emissions. The additives of interest to NAPCA were NaOH, NH_4OH, $Ca(OH)_2$ and $Mg(OH)_2$. The sulfur compounds that formed could be removed from the gas stream by conventional techniques.

Methods of incorporating each of these basic additives in the emulsions were developed. Reagent grade compounds were used in all cases.

The water-soluble additives (NaOH and NH_4OH) were first dissolved in water before emulsification with the oil.

For the water-insoluble additives ($Ca(OH)_2$ and $Mg(OH)_2$), initial attempts to produce stable emulsions were made by mixing the additive with either water or oil. In the case of $Ca(OH)_2$ it made no difference if the compound was mixed first with water or oil; stable water-in-oil emulsions resulted in either case. Conversely, it was discovered that $Mg(OH)_2$ had to be mixed with the oil first. If $Mg(OH)_2$ were wet with water first, stable emulsions could not be obtained. The emulsions were prepared by adding the aqueous phase to the oil phase in one step at room temperature and mixing for 2 to 3 minutes on the Waring Blendor.

About 10 weight percent of each additive was incorporated in emulsions containing 20 weight percent water. (Concentrated NH_4OH was used to yield emulsions containing about 7 weight percent NH_3 and 23 weight percent water.) In each case, the amount of additive was more than sufficient to react with all the sulfur present in any of the oils.

With the exception of NH_4OH, the additives were compatible with all the oils tested. Table 12 summarizes the data. It should be noted that NH_4OH formed stable emulsions with only the Mid-Continent and Gulf Coast oils.

Additional experiments with NaOH suggested that the compatible additives and water could be incorporated in the emulsions in a wide range of concentration ratios.

TABLE 12. COMPATIBILITY OF BASIC ADDITIVES AND EMULSIONS

Oil	Geographic Source of Crude	Additive			
		NaOH	NH_4OH	$Ca(OH)_2$	$Mg(OH)_2$
A	Mid-Continent	S	S*	S	S
B	West Coast	S	U	S	S
C	--	S	U	S	S
D	Africa	S	--**	S	S
E	Mid-Continent	S	S	S	S
F	Venezuela	S	U	S	S
G	Gulf Coast	S	S	S	S
H	Venezuela	S	U	S	S
J	West Coast	S	U	S	S

S = Stable

U = Unstable

* Emulsion broke after several weeks.

** Since Oil D had to be heated for processing NH_4OH was not added.

Properties of Water-in-Residual Fuel Oil Emulsions

Emulsification of residual fuel oil has a pronounced effect on some fuel properties important to combustion. Viscosity is of prime importance in its effect on atomization and on preheat required for pumping and atomization. Therefore, rather extensive measurements were made of viscosity of both the base oils and the emulsions.

Static Viscosity Measurements

The kinematic viscosity of emulsions prepared with Oil A was measured at 122 F using a Saybolt viscometer equipped with a Universal orifice. Table 13 lists the results of these measurements.

It was observed that the viscosity of the emulsions increased with water content. The presence of the water-soluble additive NaOH had little or no effect on the viscosity of the emulsion; however, the insoluble additives, $Ca(OH)_2$ and $Mg(OH)_2$, increased the viscosity of the emulsions. Air which was whipped into the emulsions during mixing, interfered with viscosity measurements, causing high values in some cases.

TABLE 13. VISCOSITY OF EMULSIONS WITH OIL A

Composition, weight percent		Viscosity at 122 F seconds, Saybolt Furol
Oil A	100	262
Oil A Water	95 5	267
Oil A Water	90 10	360
Oil A Water	80 20	385
Oil A Water	70 30	760
Oil A Water NaOH	70 20 10	442
Oil A Water $Ca(OH)_2$	70 20 10	916
Oil A Water $Mg(OH)_2$	70 20 10	1100

Dynamic Viscosity Measurement

Viscosity measurements were also made using a Ferranti-Shirley cone-plate viscometer. This instrument is useful for determining viscosity as a function of both temperature and of shear rate. For non-Newtonian materials, such as emulsions, the viscosity may vary with time as well as with shear rate due to structural changes. Because the cone-plate viscometer shears the sample continuously, time-dependent variations can be studied easily.

For these measurements, the Ferranti viscometer was programmed to shear a sample of emulsion at shear rates from zero to over 15,000 reciprocal seconds. This range should cover all of the conditions encountered in an emulsion in pipe flow. Shear rates in a burner nozzle might be momentarily higher. Measurements were made at 100 F, 122 F, and 160 F.

Viscosity-temperature data was obtained on 22 samples which included:

- 9 residual oils, as received
- 9 emulsions, one of each oil with 20-percent water
- 2 additional emulsions of Oil A containing 10 and 30-percent water, respectively
- 2 emulsions of Oil A containing 20-percent water and 10-percent $Ca(OH)_2$ and $Mg(OH)_2$, respectively.

Data were recorded continuously on a X-Y recorder, with shear rate (a function of cone speed) on the Y axis and shear stress on the X-axis. Thus, a continuous curve of shear stress versus shear rate was obtained for each sample.) Viscosity is defined as the ratio of shear stress to shear rate at any point on the curve. Three shear rates were selected for data tabulation: about 1700, 8400, and 15,000 sec^{-1}. (Actual shear rate differed slightly for specific runs because two different cones with slightly different constants were used. The variation is not significant for data interpretation.)

The viscosity calculated from the basic data is in absolute units, namely centipoises. These units may be converted to kinematic viscosity

(centistokes) by dividing the absolute viscosity by the sample density at the temperature of measurement. Kinematic viscosity may be further converted to seconds Saybolt Universal or seconds Saybolt Furol through use of ASTM tables. Since one of the objectives of this study was to observe the effects of temperature on the residual fuels and their emulsions, most of the data have been expressed in seconds Saybolt Universal, (SSU) and plotted on ASTM D341-43 graph paper. The Saybolt Furol scale applies only to measurements at 122 F and hence cannot be used for showing viscosity-temperature relationships. However, Saybolt Furol values were calculated for all emulsions at 122 F and are also reported. The Furol values obtained were in reasonable agreement with the values available from suppliers of the fuels.

The absolute accuracy of these data are affected by a number of possible errors, as will be discussed. Overall, the cumulative errors are not highly significant, in most cases being less than 10 percent. Thus, the data as given are quite adequate for both a general impression of emulsion behavior and for engineering calculations of pipe flow.

One source of error is shear heating. At the viscosity level of these fuels and their emulsions, even the excellent temperature control features of the Ferranti viscometer cannot prevent shear heating at the higher shear rates. The amount of shear heating for each determination was registered by a thermocouple in the plate of the viscometer. At the highest viscosities and shear rates (at 100 F), this rise was typically four or five degrees F. At the lowest viscosity tabulated at a shear rate of 1700 sec^{-1}, the rise was negligible. The rate of change of viscosity with temperature is about 4 percent per degree at 100 F for these fuels. Therefore, a five degree rise in temperature is significant. This rise was taken into account in interpretation of the flow curves for non-Newtonian character. All of them display what could be interpreted as reversible shear-thinning, but calculation showed that most of that could be accounted for by shear heating. It is for this reason that the decision was made to ignore non-Newtonian shear thinning in plotting the results.

TABLE 14. VISCOSITIES OF RESIDUAL OILS AND EMULSIONS
(At Shear Rate of 1700 sec^{-1})

Residual Oil Sample	Viscosity, seconds Saybolt Universal, at 122 F		Viscosity, seconds Saybolt Furol, at 122 F	
	Base Oil	20-percent water emulsion	Base Oil	20-percent water emulsion
Oil A	1850	3020	188	307
Oil B	1950	1850	198	188
Oil C	2240	3600	228	367
Oil D	487	779	51	80
Oil E	1980	2240	202	228
Oil F	2100	2920	214	297
Oil G	1730	2760	177	281
Oil H	1790	2760	182	281
Oil J	1330	3330	135	339

Emulsion	Viscosity, seconds Saybolt Universal, at 122 F	Viscosity, seconds Saybolt Furol, at 122 F
Oil A+10-percent water emulsion	2340	238
Oil A+30-percent water emulsion	4570	470
Oil A+20-percent water +10-percent $Ca(OH)_2$ emulsion	3600	367
Oil A+20-percent water +10-percent $Mg(OH)_2$ emulsion	2730	277

A second source of errors in the data is the decision to ignore density in converting centipoises to Saybolt units. Enough data were available for the samples to show that their specific gravity were mostly in the range from 0.95 to 1.0 at 75 F. The decrease in density for hydrocarbons of this type between 75 F and 160 F is only about 4 percent. Therefore, because laboratory determination of actual densities at elevated temperatures for the samples would have required extensive effort, it seemed reasonable to ignore the small error involved.

A third source of error in the viscosity values is the small error caused by time-dependent viscosity changes in the emulsion samples. Most of the emulsions experienced a small change of viscosity due to shear during the measurement procedure. However, this viscosity change was usually only a few percent. The few cases of gross change are noted in the graphs or the discussion of results given later.

Measurements showed that the residual fuels themselves seem to be slightly shear-thinning at high shear rates (10,000 sec^{-1} or more). Addition of emulsified water increases this tendency for some of the oils. However, the maximum temporary viscosity loss is on the order of 20 percent. Therefore, this amount of non-Newtonian behavior does not seem highly significant in terms of pipe flow or atomization.

Finally, there may have been some slight error in the determinations made at 160 F because of evaporation of part of the emulsion. Nearly all the samples, including the fuels themselves, showed a tendency to increase in viscosity with time. The most probable explanation for this is evaporation of a small amount of light hydrocarbon.

Overall, despite the errors noted, the results are believed to be reasonably accurate for the purpose intended.

Viscosities of Emulsions Compared to Base Fuels

Figures 6 to 15 present results of the viscosity measurements. These plots of viscosity in seconds, Saybolt Universal, versus temperature have been arranged to provide a comparison between the particular residual fuel sample and its emulsion. Saybolt Universal values are approximately ten times as large as Saybolt Furol values. Table 14 summarizes the viscosity data in both units.

The data show that in all cases, with the possible exception of Oil B, the emulsion is more viscous than the base fuel by amounts ranging from 50 to 100 percent. The difference is relatively small for Oils B and E and reaches a maximum with Oil J. The viscosity-temperature dependence of the emulsions is little different from that of the fuels.

Figure 6 shows the viscosities of emulsions of Oil A with 10, 20, and 30 percent water. The data show that the viscosity of the emulsion increased with increasing water content.

The special emulsions of Oil A were examined in which solid material had been dispersed. One contained calcium hydroxide, and the other contained magnesium hydroxide. The calcium hydroxide emulsion was considerably higher in viscosity than the magnesium hydroxide one (Fig. 15), and it was unstable at the highest shear rates at 100 and 122 F. The magnesium hydroxide emulsion showed evidence in the storage bottle of having some free water or at least an oil-in-water exterior phase. The bottle surface appeared to be water-wet rather than oil-wet as for all the other emulsions.

Flow Instabilities Observed in Emulsions

Three types of instability were observed during the viscosity measurements. One was a time-dependent shear-thickening. That is, the viscosity increased slightly due to the shearing action generated during the viscosity determination itself. This was observed for nearly all of the emulsions at all temperatures. A likely explanation is that the emulsions had begun to grow coarser in texture due to long standing. Thus the shearing had the effect of homogenizing or redispersing the emulsion to a finer texture.

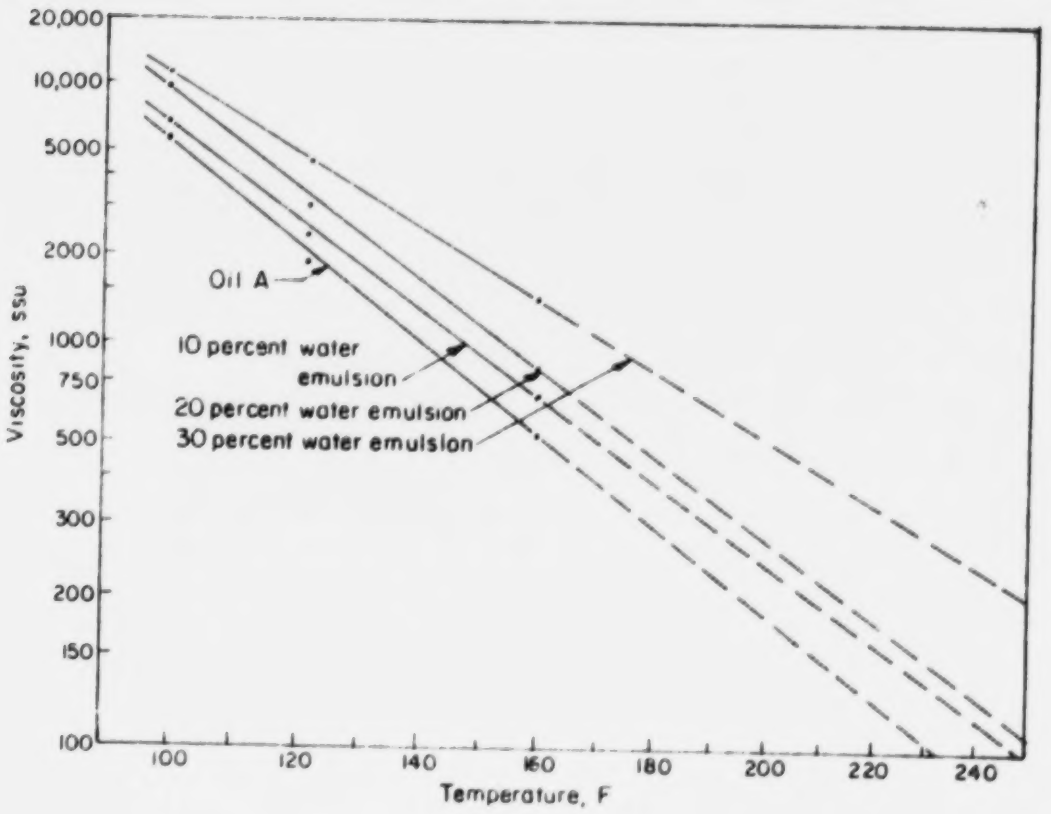

FIGURE 6. VISCOSITIES OF OIL A AND WATER-IN-OIL A EMULSIONS

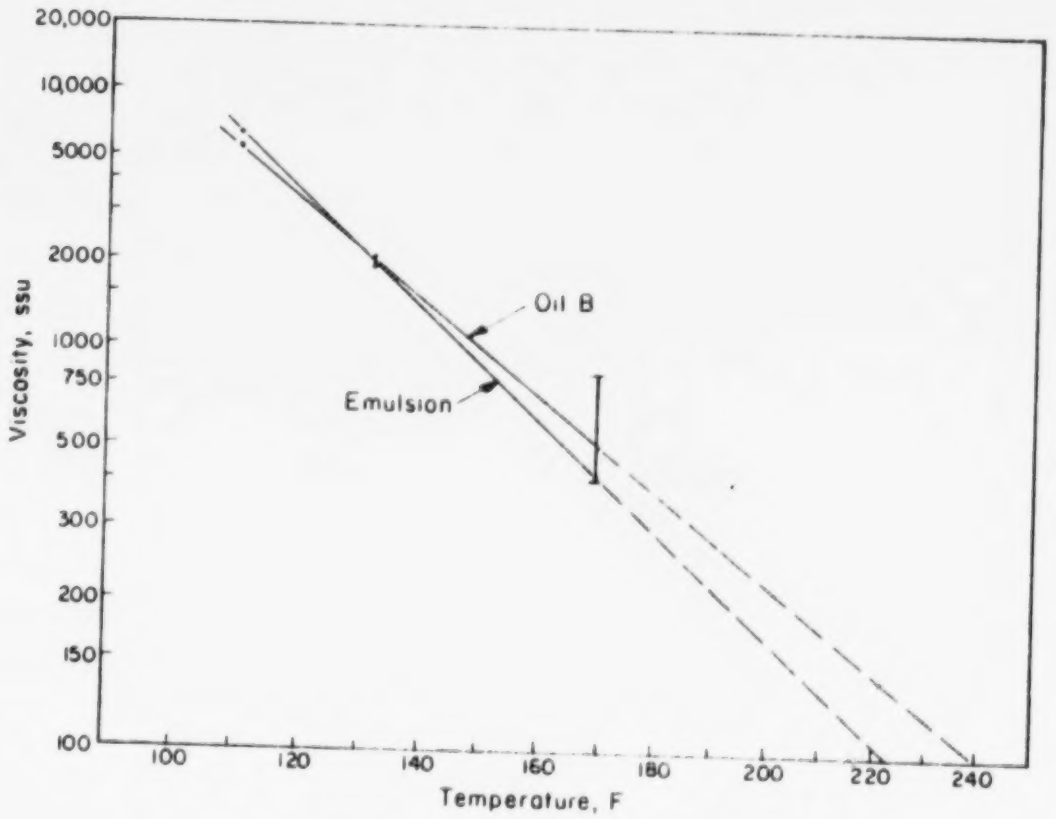

FIGURE 7 VISCOSITIES OF OIL B AND WATER-IN-OIL B EMULSION

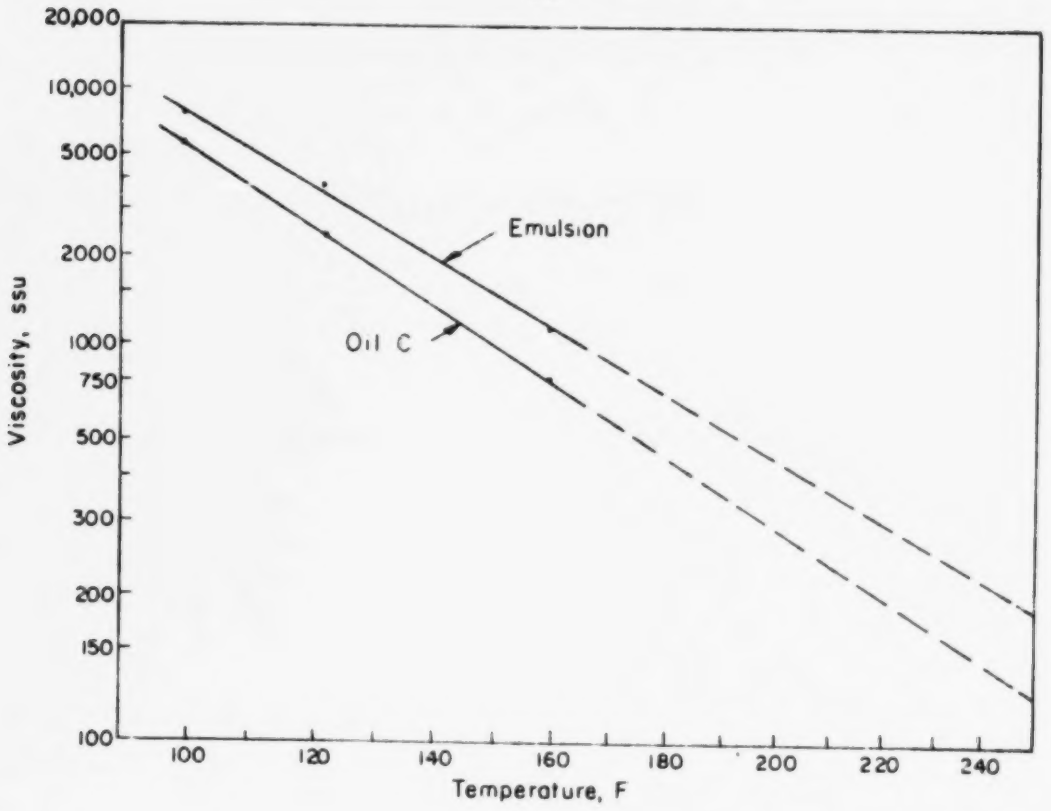

FIGURE 8. VISCOSITIES OF OIL C AND WATER-IN-OIL C EMULSIONS

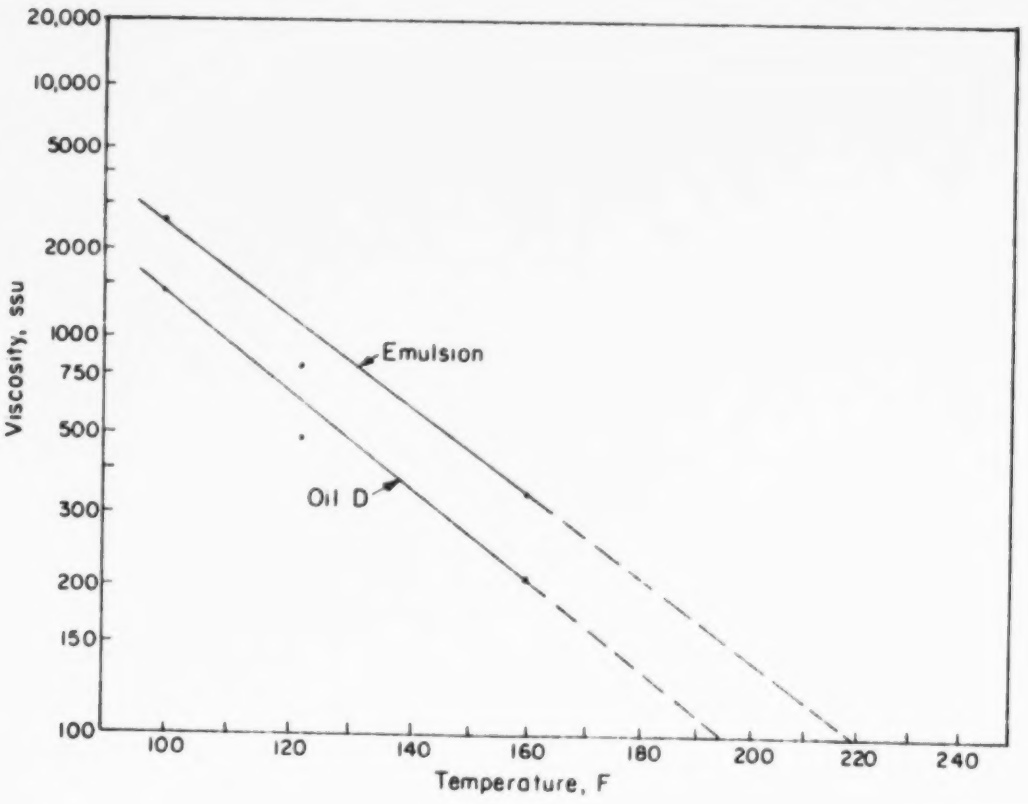

FIGURE 9 VISCOSITIES OF OIL D AND WATER-IN-OIL D EMULSIONS

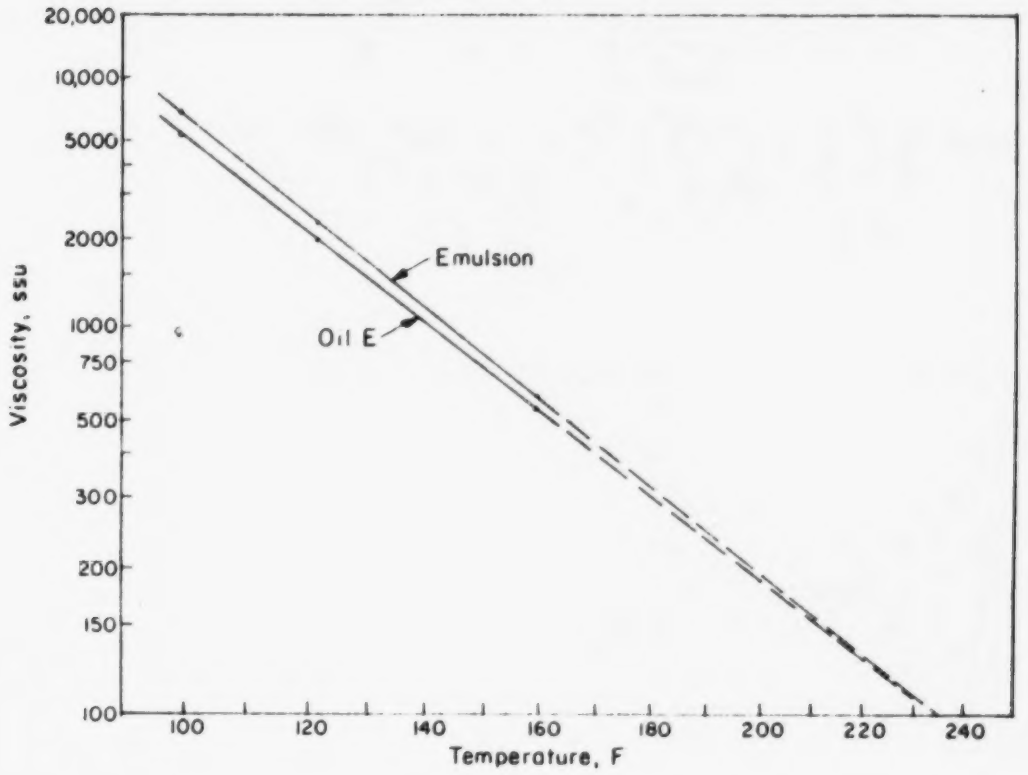

FIGURE 10. VISCOSITIES OF OIL E AND WATER-IN-OIL E EMULSION

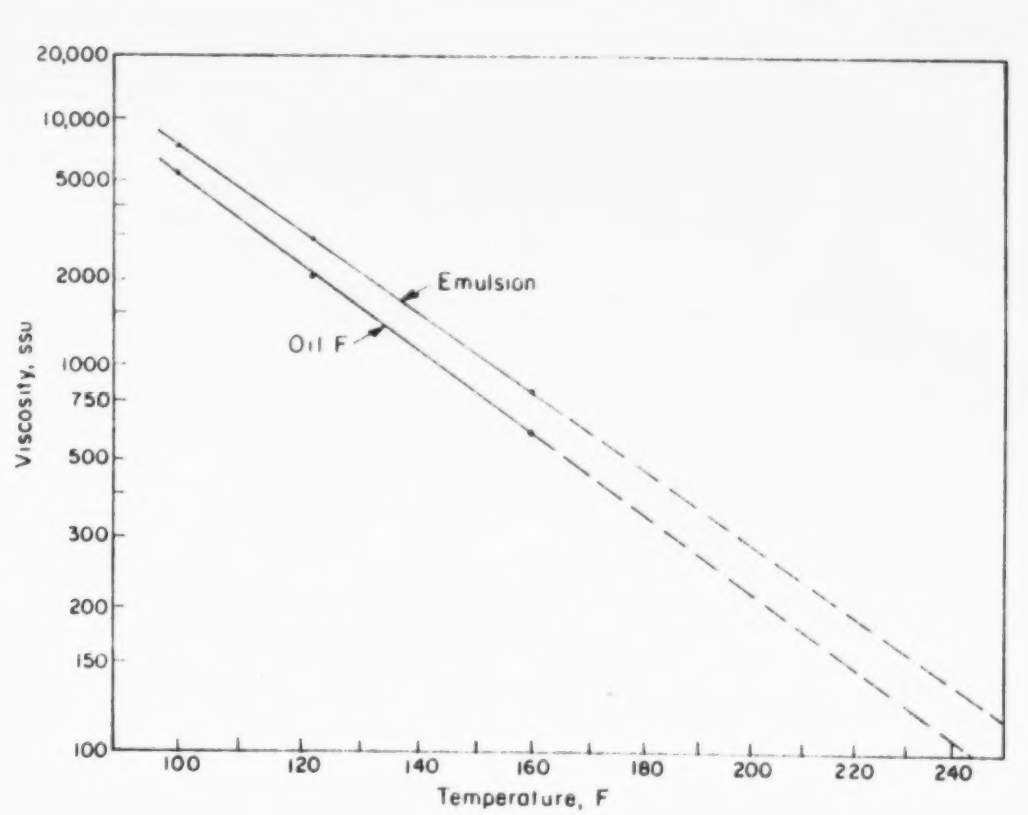

FIGURE 11 VISCOSITIES OF OIL F AND WATER-IN-OIL F EMULSIONS

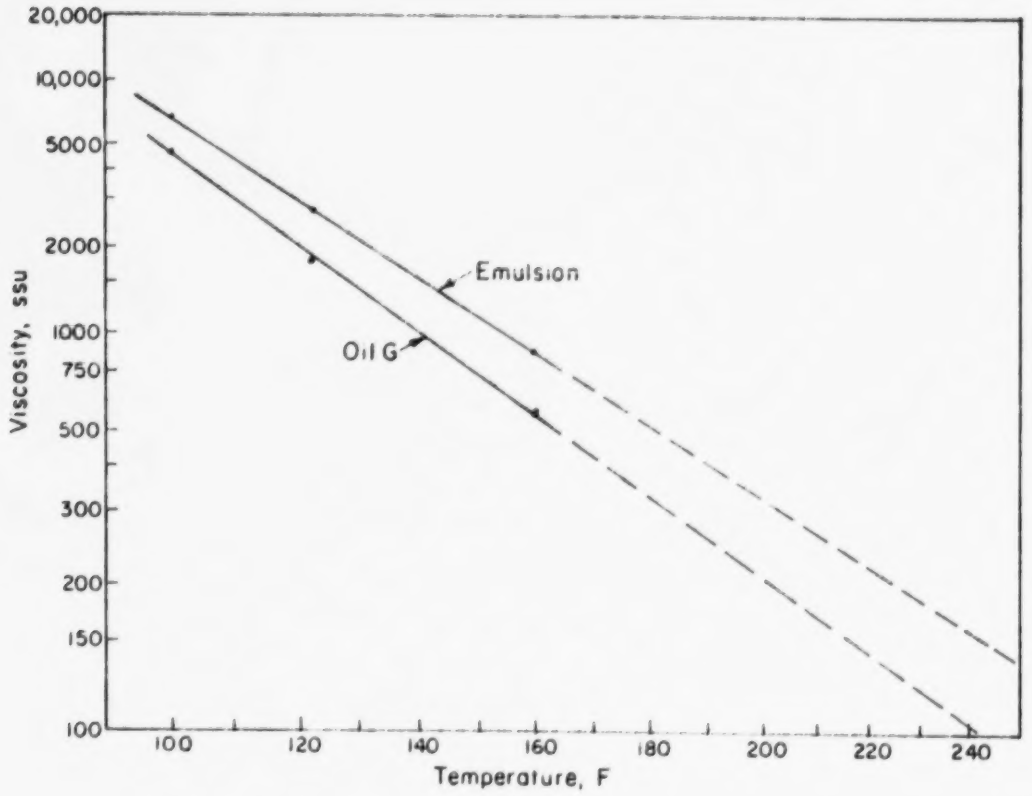

FIGURE 12. VISCOSITIES OF OIL G AND WATER-IN-OIL G EMULSIONS

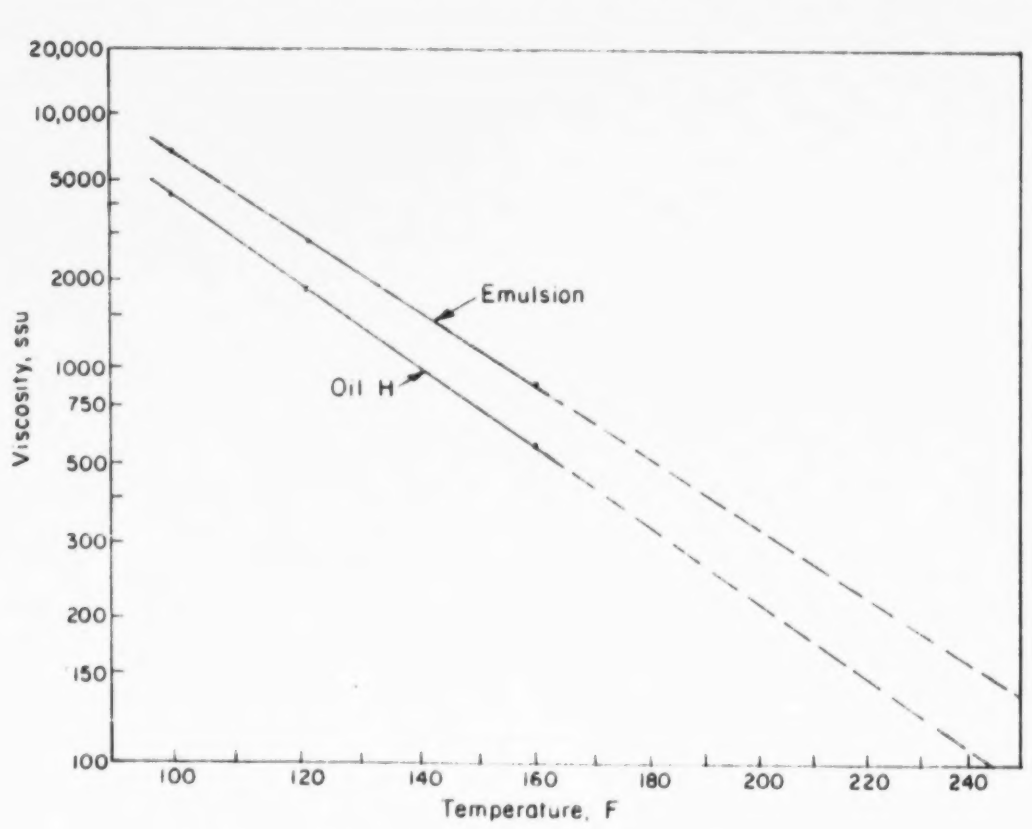

FIGURE 13. VISCOSITIES OF OIL H AND WATER-IN-OIL H EMULSIONS

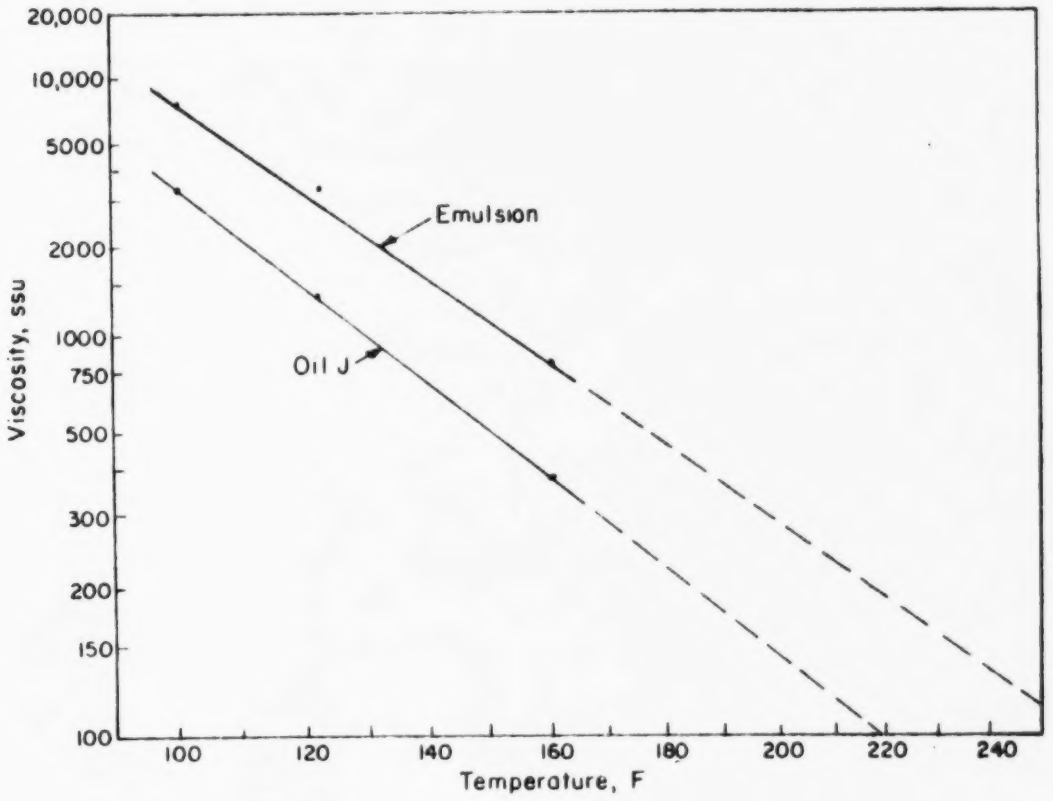

FIGURE 14. VISCOSITIES OF OIL J AND WATER-IN-OIL J EMULSIONS

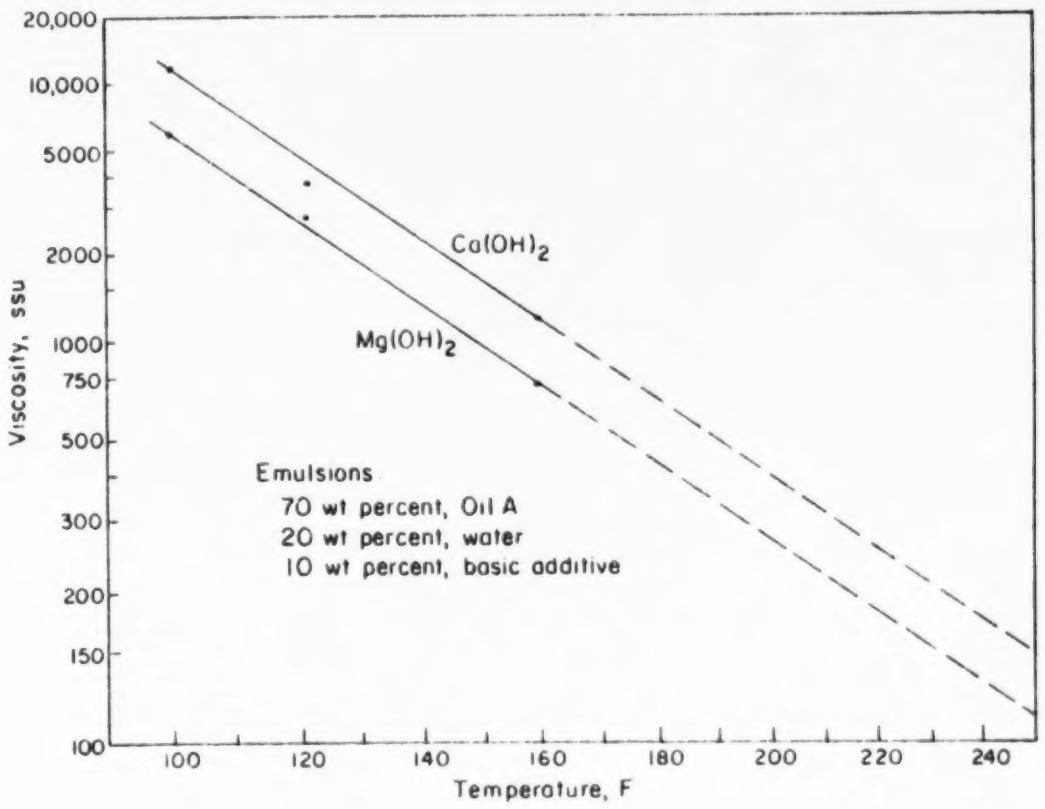

FIGURE 15. VISCOSITIES OF WATER-IN-OIL A EMULSIONS CONTAINING BASIC ADDITIVES

The second type of flow instability observed is known to rheologists as the "Weissenberg Effect[29,30]. In the cone-plate viscometer, the phenomenon appears as an escape of the sample from the shear zone between the cone and plate. Some of the fluid actually flows out of the shear zone and climbs up on the revolving cone. The effect may be seen visually before an actual loss of fluid from the shear zone occurs because the meniscus around the edge of the shear zone becomes doughnut shaped. When fluid actually leaves the shear zone, the torque (shear stress) on the cone drops suddently and erratically. If the shear rate is lowered, the fluid will usually relax and refill the shear zone. At present there is no satisfactory explanation of why and how this phenomenon occurs in emulsions, it occurred only with the Oil A emulsion containing 30 percent water at 100 and 122 F, and with the Oil B emulsion at 122 F. The Oil B emulsion also showed signs of such behavior at 160 F, but with other complications.

The structure which gives rise to the Weissenberg effect probably would not significantly alter flow of the emulsion in pipes. However, it might possibly alter the spray pattern generated from a nozzle.

The third type of flow instability was a partial separation or breaking of the emultion to release free water. This phenomenon was observed with the following:

Oil B emulsion at 122 F
Oil D emulsion at 160 F
Oil C emulsion at 160 F
Oil H emulsion at 160 F.

Free water could be seen on the surface of these samples after shearing when the cone and plate were separated.

It is not clear what effect these instabilities might have on the behavior of these emulsions in handling in a burner system. Actual pumping and burning tests would be necessary to determine what effects, if any, occur.

Thermal Stability Tests

The thermal stability of the emulsions were investigated at both high and low temperatures. At elevated temperatures, NH_4OH is readily lost from the emulsion due to evaporation. However, all the other emulsions were found to be stable for at least three days at 160 F. In addition, all emulsions (except that containing NH_4OH) were placed in a boiling water bath (212 F) for one hour. Except for a slight loss of water by evaporation, no changes were apparent. Finally, no changes were apparent when the emulsions were stored at 6 F for 2 hours. Thus, each of the emulsions except those containing NH_4OH appear to be stable over a wide temperature range.

Atomization Trials

Atomization trials were conducted without combustion to determine if the water-residual fuel oil emulsion remained emulsified or broke due to shear stresses during the atomization process. The atomization trials were conducted using a 20 percent water emulsion of Oil A. A quantity of base Oil A was also atomized to provide a reference for comparison purposes.

To lower the viscosity for pumping, a double-boiler arrangement was used to preheat the oil to a temperature of 130 F to 165 F. A steam jacket was used over the fuel line to heat the fuel to the desired temperature for atomization. Due to the fact that the water in the emulsion would readily evaporate or boil at high temperatures (200 F to 250 F), the steam jacket was located downstream of the pump; the pressure in this section of the fuel/line would prevent boiling. The emulsion was atomized at temperatures of 175 to 250 F. Base oil was atomized at 115 to 200 F.

A Delavan"Industrial Aero" air-atomizing nozzle was used for these trials. Both the atomizing air pressure and the fuel line pressure were maintained at about 20 psig. The fuel flow rate was about 1.1 to 1.3 gph.

Samples of spray drops were collected during atomization of both the water-residual oil emulsion and the residual oil. Drops were collected on clean glass microscope slides by manually passing the slides through the spray. Individual drops were examined under 200X magnification.

Examination of emulsion drops showed a multitude of small (several micron) circles within the drops which were interpreted to be water droplets within the emulsion drop. However, a smaller number of similar circles were also observed near the surface of drops of the residual oil. Existence of these circles in the residual oil caused some uncertainty in identifying the circles as water droplets. This prevents a positive statement that water droplets were contained in the emulsion drops on the slides. However, no separate water and oil phases were observed on the slides, suggesting that the emulsion had not broken.

For the emulsion only, an estimate was made of droplet sizes from the droplets collected on the slides. These estimates are:

Emulsion temperature at nozzle, F	Estimated droplet diameter range, μ	Estimated number-average droplet diameter, μ
175	4-250	100
200	4-200	100
250	4-100	60

From observations of the sprays it appeared that atomization of the emulsion at 250 F produced a spray fineness that approached the fineness of a distillate oil spray at room temperature. Atomization of emulsion at 175 F and 200 F produced coarser sprays. It was not possible to determine if "secondary atomization" of the emulsion droplets might have been occuring at 250 F although this may have been happening as the fuel temperature was above the boiling point of water.

Results of these "cold" atomization trials were interpreted to show that residual oil-water emulsion can be atomized in conventional fuel systems without the emulsion breaking. The higher fuel viscosity will probably require

higher preheat temperatures. It was decided that the only real measure of the atomization quality of the emulsion will be results obtained in a combustion rig.

PRACTICAL CONSIDERATIONS IN UTILIZING EMULSIFIED FUELS

A number of practical problems must be evaluated when considering the use of emulsified fuels. The previous report[3] mentioned some of these problems, including:

(1) Logistics (Either stable emulsions or on-site preparation are required and, for premixed emulsions, the water adds to the total weight to be transported)

(2) Corrosion (The presence of water in the fuel system is likely to aggrevate corrosion problems)

(3) Stability (Long-term storage stability is needed at a range of temperatures.)

Two additional problems come to mind when considering the use of emulsified residual fuels, including emulsions containing basic additives. These two problems are:

(4) Pumpability and atomization problems

(5) Corrosion and deposits on the boiler fire-side surfaces due to the presence of basic additives.

Each of these problems will be discussed in the following report sections.

Other factors to be recognized in the use of emulsified fuels include a slight reduction in theoretical flame temperature and a slight increase in heat loss. Both of these effects are caused by the heat absorbed by the water in the emulsion and were explored in the previous report.

Pumping and Atomization

Data in the technical literature[11] and results of Battelle's experimental studies both indicate that the viscosity of water-in-oil emulsions is greater than the viscosity of the base oil. Because the viscosity of residual

oils is already high, preheating is required for pumping and atomization. Thus, an increase in viscosity presents a practical problem which must be evaluated. In fact, the detailed viscosity measurements reported earlier were designed to provide the input for this evaluation.

Figure 16, from Siegmund[27], shows the limits of easy pumpability and the absolute limit of pumpability. He states that, "above 1000 cp (about 4600 SSU) pumping difficulty can be expected and above 2500 cp (about 11,000 SSU) a major loss in pump capacity is normally encountered". This indicates the level of preheat required for handling of No. 6 oils.

For atomization, No. 6 fuel oils are usually preheated to the temperature required to reduce viscosity to 100 to 150 SSU[27,31,32]. Oils in this viscosity range can be atomized in pressure or two-fluid atomizers conventionally used.

Figure 17 shows the approximate temperature required to reduce the viscosity of No. 6 oils to 150 SSU. Because most No. 6 oils are in the 130 to 210 seconds, Saybolt Furol, viscosity range at 122 F (1300 to 2000 seconds, Saybolt Universal, at 122 F) the normal fuel temperature for atomization is 204 F to 218 F.

However, the viscosity increase of 20 percent water emulsions for the nine oils utilized in this study would require an average of about 20 F additional preheating to produce the same viscosity for atomization. (The better atomization expected of emulsions due to "secondary atomization" may reduce the required fuel temperature slightly, but this defeats one purpose of firing the emulsion.)

Heating of emulsions to temperatures above 160 F for long periods of time may tend to evaporate the water and destroy the emulsion. Therefore, for firing of emulsions it appears that the final heating of the oil must occur between the fuel pump and the nozzle. At this location, the fuel is under pressure and evaporation and/or boiling will be reduced.

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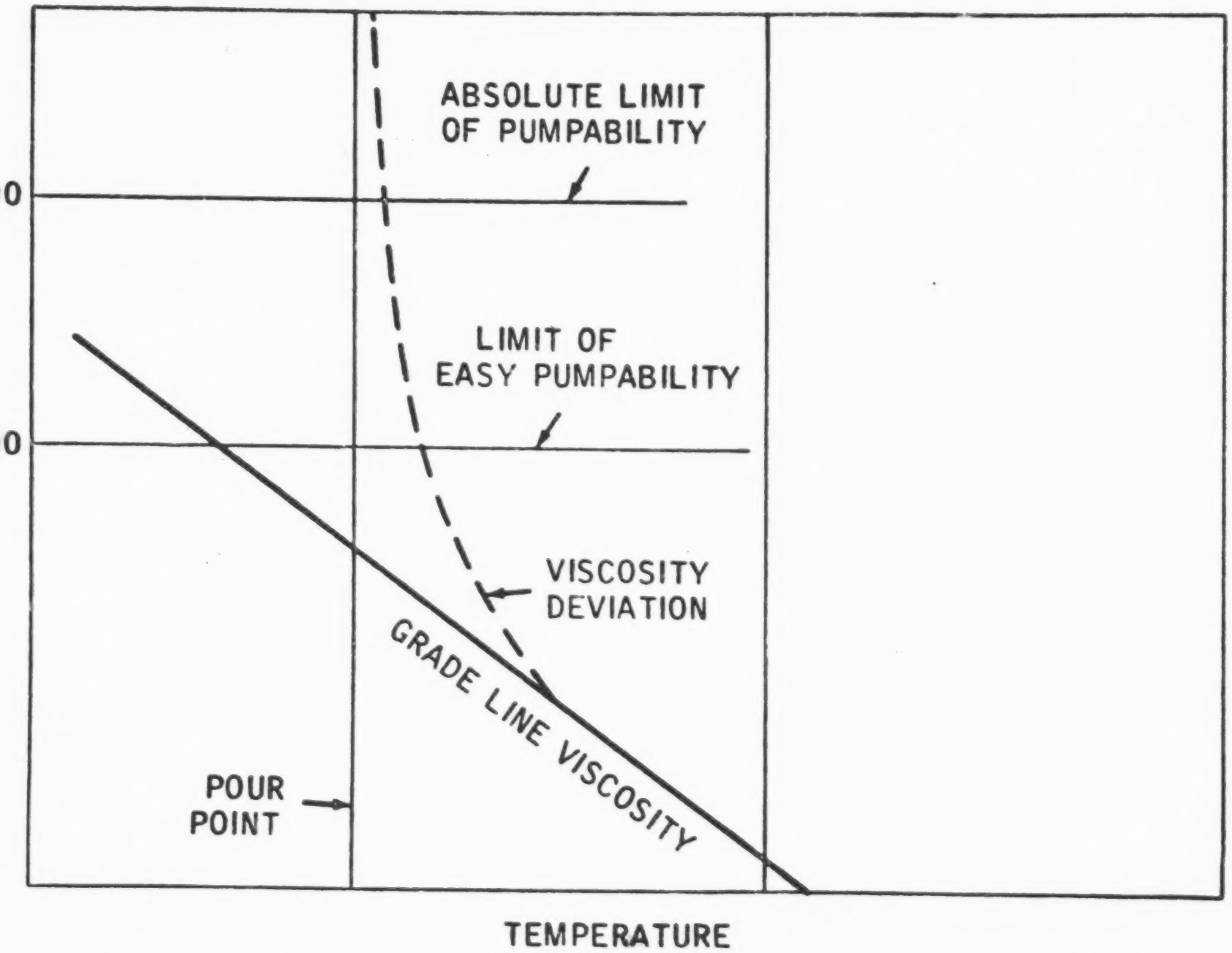


FIGURE 16. DEVIATION OF VISCOSITY FROM GRADE LINE ABOVE POUR POINT [27]

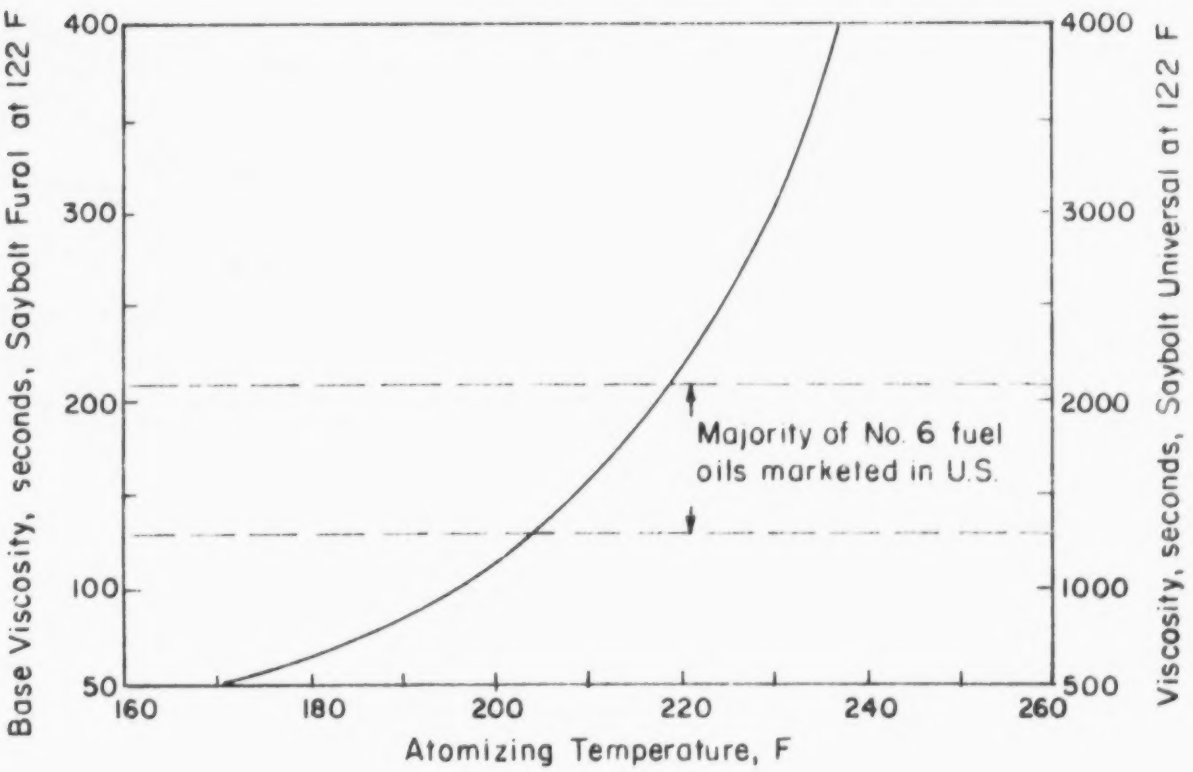


FIGURE 17. PREHEAT NEEDED FOR RESIDUAL OILS HAVING DIFFERENT BASE VISCOSITIES TO REDUCE THE ATOMIZING VISCOSITY TO 150 SSU

Therefore, if heating of the fuel is required for pumping, also likely due to increased viscosity, two separate heating systems will be required. This may present an unacceptable burden to the operator.

However, Ludera[33] claims that water-in-oil emulsions can be produced that have lower viscosities than the base oils. This reference should receive further attention.

Effect on Corrosion and Deposits of Basic Materials in Fuel Oil-Water Emulsions

Alkaline materials may be added to residual fuel oil-water emulsions to capture SO_2 resulting from combustion of sulfur-bearing fuels. If the resulting change in pH of the water phase does not affect the characteristics of the emulsifying surfactant, the SO_2-capturing additive can be any of several different substances. Practically, if any appreciable amount of SO_2 were to be removed, the additive must be inexpensive, which practically limits it to slaked lime or to calcined dolomite or magnesite. Other factors aside, NaOH or Na_2CO_3 might be considered as alkaline additives but their cost would be excessive in the amounts required. For example, if the residual fuel contained 4 percent sulfur, or roughly 13.5 pounds of sulfur per barrel of fuel oil, then 27 pounds of SO_2 would be formed and approximately 24 pounds of CaO or 17 pounds of MgO would be necessary on a stoichiometric basis to convert the SO_2 from one barrel of fuel oil into $CaSO_4$ or $MgSO_4$. If NaOH were added in sufficient amounts to convert all the SO_2 in the flue gas to Na_2SO_4, about 34 pounds would be required per barrel of fuel oil. The question then arises, what effect will these large amounts of additive have on corrosion and deposits in the boiler?

Residual fuel oil seldom contains more than 0.1 percent ash, with the value usually about 0.05 percent. Even for the higher level, the ash in a barrel of fuel oil would weigh only 3.4 pounds, so that the amount of CaO to be added to capture all the SO_2 from that barrel of oil would be 7 times greater than the ash in the fuel. With MgO, the weight of additive would be 5 times greater than the ash, and with NaOH, 10 times greater. Therefore, it is evident

that the solids in the flue gas would be predominantly from the additive, and that the ash in the fuel would be an insignificant part of any deposits that formed on heat-receiving surfaces.

Corrosion of superheaters and reheaters in oil-fired steam generators invariably can be attributed to the formation of a liquid phase on the metal surface at temperatures in the range of 1100 F to 1300 F. These highly corrosive liquid films are of two types, (1) alkali iron trisulfates such as $Na_3Fe(SO_4)_3$, and (2) vanadium compounds, typically sodium vanadyl vanadate, $5Na_2O \cdot V_2 \cdot 11V_2O_5$ (melting point of 1071 F), and vanadium pentoxide, V_2O_5 (melting point of 1243 F). The corrosion mechanisms here are not important, but it has been thoroughly established that corrosion will not take place unless a liquid phase is present; gas-phase oxidation is no worse in flue-gas atmospheres than in air so that it can be essentially ignored for alloys normally suitable for high-temperature operation. Liquid films, however, are highly objectionable and must be prevented.

Two methods exist for controlling such liquid films: (1) establish conditions that prevent the series of chemical reactions that lead to the formation of these low-melting compounds, or (2) provide sufficient nonreactive material to absorb the molten compound physically so that it has no access to the metal surface below the deposit. The large amounts of CaO or MgO recessary to capture SO_2 would function by both of these methods to prevent metal wastage. Lime and magnesia are highly reactive with SO_3, and in their presence the SO_3 in flue gas will be vanishingly small, certainly less than 5 ppm. Under such conditions, the complex trisulfates cannot form and external corrosion would not occur by reaction between the alkalies in the oil ash and the metal surface. These reactions can only take place at superheater metal temperatures when the SO_3 exceeds about 250 ppm. Sulfur trioxide plays no part in the formation of vanadium compounds, but here the physical adsorptive nature of the lime and magnesia would prevent the presence of a continuous liquid film.

Added sodium salts would behave oppositely. The compound that would form eventually is Na_2SO_4. Its melting point is 1625 F, so that the direct reaction product, by itself, would not be molten at superheater metal temperature. However, the tendency to form the molten complex trisulfates at the expense of the oxides on the hot metal surfaces would be greatly enhances with large amounts of Na_2SO_4 present. Also, the formation of low-melting sodium vanadates could be expected. Hence, increasing the sodium content of deposits would add greatly to the likelihood of serious metal wastage. Therefore, no sodium compounds should be added to fuel oil-water emulsions if metal wastage is to be prevented.

It can be concluded, then, that enough lime and magnesia added to a fuel oil-water emulsion to capture the SO_2 in flue gas would provide conditions on superheater and reheater surfaces such that no metal wastage should occur. Low-temperature corrosion of air heaters similarly would be unlikely because no SO_3 would be present in the flue gas and hence the dewpoint would be low.

Formation of deposits could be troublesome with additives. Coals containing large amounts of CaO in their ash, for example Australian brown coals, frequently cause trouble by forming massive deposits of $CaSO_4$ on heat-receiving surfaces. This effect usually is attributed to the nature of the $CaSO_4$ crystallites, which develop an interlocking structure that is extremely difficult to disrupt with soot blowers. Magnesia may behave similarly, but its role is less clear. Experience thus far with limestone injection systems for capturing SO_2 when burning pulverized coal has not shown similar problems with deposits, but operating periods have been short and this point has not yet been proved. Sodium compounds invariably lead to objectionable deposits, and it has been shown repeatedly with coal ash that the amount of deposit formed is directly related to the sodium content of the deposit.

Other alkaline additives than NaOH, CaO, or MgO, for example ammonia or methyl amine, or any other additive not containing a metal salt or compound,

will have no effect on corrosion and deposits. Additives have not been shown to affect the combustion process enough to influence formation of SO_3 in flames. Similarly, no additives free of metal ions have been shown to have any effect, either positive or negative, on corrosion or deposits.

Results of trials of steam injection into the combustion region of a laboratory combustion apparatus have shown that the presence of the increased moisture tended to reduce the rate of deposit formation[34]. Water in the fuel in the form of an emulsion might behave similarly.

In summary, additives containing sodium should be avoided as they are likely to create corrosion and deposit problems. Other additives should not present these problems.

PRELIMINARY DESIGN OF A COMBUSTION TEST RIG

The final aspect of this study consisted of the design of a laboratory-scale combustion unit for simulation of industrial combustion processes. This small burner-furnace rig suitable for burning residual fuel, emulsified fuel, and other liquid fuels at about 2 gph, is intended to simulate combustion conditions comparable to those in industrial boilers. This rig may be used for studies of factors affecting combustion quality and pollutant emissions such as: additives, emulsions, and various combustion parameters including recirculation and two-stage burning.

Special consideration in the design of the combustion rig was given to aspects which result from the large scaling factor (about 600 to 1). When scaling combustion processes over this range, it is impossible to simultaneously maintain both combustion-zone residence time and turbulence in the model equal to those of the full-scale unit. However, to reduce combustion volume and conserve space, industrial units are designed for higher turbulence levels than are necessary for satisfactory combustion. Therefore, it appears possible to achieve the desired residence times while maintaining satisfactory combustion, although the turbulence level would be below that of industrial units.

A combustion rig designed to duplicate the residence times of a full-scale industrial unit can also be operated at high turbulence levels by restricting the inlet air passage to increase the pressure drop.

Therefore, a combustion rig was designed for maximum flexibility of operation, so as to permit conducting experiments at both high turbulence and at long residence times. Flexibility can be provided by the use of variable geometry air registers, variable swirl, variable air and fuel pressures at atomizers, variable firing rate and air-fuel ratio, and by controlling wall temperatures with insulation and/or auxiliary heating.

Design of the unit also included simulation of the time-temperature profile of industrial units in the post-combustion region.

Characteristics of Industrial Boilers

The objective of the preliminary design of the combustion test rig is to simulate as nearly as possible the furnace conditions existing in full-scale industrial boilers generating about 100,000 lb/hr of steam.

Because a boiler is designed to generate steam, the important design criteria must include providing sufficient heat transfer surface to generate the required quantity of steam. A portion of the needed heat-transfer surface is located on the furnace walls in the form of water tubes. The remaining heat transfer surface is located in the path of the flue gas leaving the furnace and takes the form of superheaters, reheaters, and economizers. The heat-transfer surface on the furnace walls must be sufficient to lower exit gas temperatures to the point where superheat temperature can be held to an acceptable level. Factors which affect the heat-transfer rate to the walls and, therefore, surface area requirements include gas temperatures and velocities, flame dimensions which influence radiation, and ash deposits on the wall tubes.

Providing a geometric arrangement of the furnace which includes the required heat-transfer surface results in a furnace volume which serves as the combustion space. The combustion intensity (in terms of heat-release rate per unit volume of furnace) for a given heat-transfer rate, is dependent upon furnace shape and size. For example, for a cubical furnace with each dimension equal to L', the wall surface area is $6(L')^2$ and the furnace volume is $(L')^3$. Therefore, the surface-to-volume ratio reduces to $6/L'$. (For furnaces of other shapes, the constant would be some value other than 6.) If heat-transfer rates are constant, the combustion intensity is a constant divided by L', and decreases as boiler size increases. Thus, for furnaces of similar shape for a given heat-transfer rate, the combustion intensity is inversely proportional to boiler size. Residence time for combustion for these similar units is inversely proportional to combustion intensity and, therefore, is directly proportional to furnace size.

In practice, an effort is made to minimize the volume of large furnaces by adding heat-transfer surface in the form of curtain walls or plattens of tubes, or by use of multiple furnaces of smaller size. Using these techniques, and by taking advantage of the higher radiation flux in larger furnaces, it has proven possible to operate with combustion intensities of 20,000 to 30,000 Btu/ft^3-hr in the largest furnaces. Although ingenuity is required to add heat-transfer capability to large furnaces, the design of small furnaces may require less than full water cooling to keep gas temperatures in the furnace high because the surface-to-volume ratio for a small furnace may be as much as 10 times that in a large furnace. Techniques for maintaining high gas temperatures include use of tube-and-tile walls (water tubes fairly widely spaced in front of a refractory wall), or by use of refractory sections in the furnace.

For the purpose of modeling a typical industrial furnace, a value of 30,000 Btu/ft^3-hr has been selected as a suitable combustion intensity. A furnace for generating steam at a rate of 100,000 lb/hr would have a firing rate of about 140,000,000 Btu/hr. Therefore, the volume of the furnace would be 4650 ft^3 or about a 17-foot cube. The unit will burn about 940 gph of No. 6 fuel oil. Surface area of this furnace would be 5 $(L')^2$ = 1650 ft^2. The average rate of wall heat transfer to provide a 2200 F furnace exit gas temperature would be about 23,000 Btu/ft^2-hr.

Time-Temperature Relationships

Little data giving gas residence time-temperature relationships in industrial boilers appears to have been published. However, residence time can be estimated by a dimensional analysis of a boiler furnace. Roughly, 10 lb/hr of air is required for combustion to generate 10,000 Btu/hr. Therefore, 30•V lb/hr air is required to support combustion at intensity 30,000 Btu/ft^3-hr in a furnace volume of V_o ft^3. The specific volume of combustion products, v_a, at an average furnace temperature of 2500 F is:

$$v_a = \frac{RT_a}{P} = \frac{53.35 \times 2960}{14.7 \times 144} = 75 \text{ ft}^3/\text{lb}.$$

Therefore, the volume of gas passing through the furnace, Q, is

$$Q = 75 \times 30 \cdot V_o \text{ ft}^3/\text{hr} = .625\ V_o \text{ ft}^3/\text{sec}.$$

Residence time in the furnace will be

$$T_o = \frac{V_o \text{ ft}^3}{Q \text{ ft}^3/\text{sec}} = \frac{V_o}{.625\ V_o} = 1.6 \text{ seconds}.$$

Likewise, residence times for other combustion intensities are:

Combustion intensity, Btu/ft^3-hr	Residence times, sec
50,000	1.0
20,000	2.4
15,000	3.2

Figure 18 shows time-temperature plots for 3 oil-fired central-station utility boilers[35]. These units are designed for firing rates of 3 to 20 times that of a 100,000 lb/hr industrial boiler. Gas temperatures at the inlet to the superheater would be 2000 F to 2300 F. Therefore, most of the residence time shown on Figure 18 occurs in the heat-transfer region and only 1.5 to 3 seconds of this time occurs in the combustion region.

Although it would be desirable to determine the absolute values for each of the various parameters for industrial furnaces, as mentioned earlier, there is little data available in the literature. However, it will be shown in the next section of this report that the scaling can be done on a comparative basis and absolute magnitudes are not needed for design purposes.

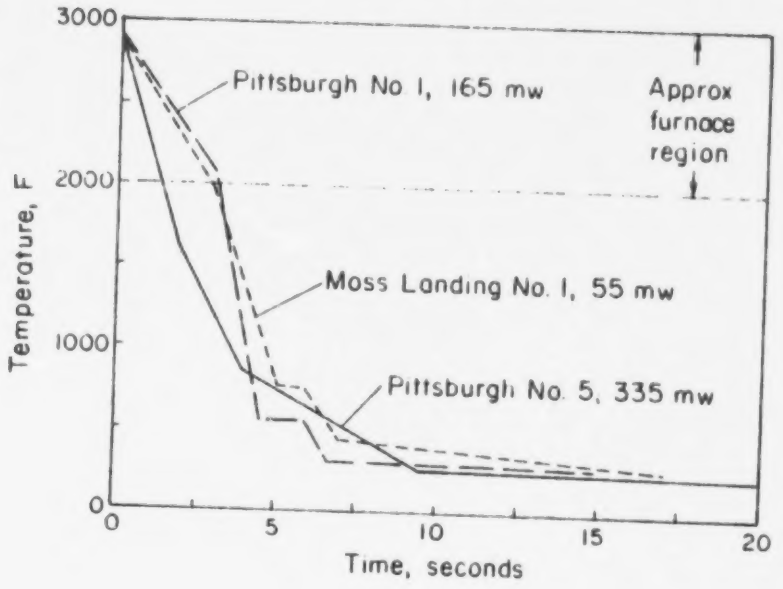


FIGURE 18. TIME-TEMPERATURE DATA FOR THREE UTILITY BOILER FURNACES

Scaling and Modeling Considerations

It is not possible to make a small-scale model that simultaneously simulates turbulence and mixing conditions and residence times. Instead, it is necessary to limit the modeling to simulate either turbulence and mixing or residence time. Both can be achieved in the same combustion rig for different runs by choice of the particular operating conditions.

Where flame geometry, flame temperature, completeness of combustion, and effects of turbulence and mixing on completeness of combustion are the variables of primary interest, it appears desirable to model with velocity and pressure drop at about the same levels as in the prototype. However, where a phenomenon controlled by chemical kinetics is of primary interest, it appears necessary to model to provide suitable time-temperature profiles rather than maintaining velocity and turbulence levels. Phenomena in combustion of heavy oil in industrial furnaces that are controlled by reaction kinetics include the formation of nitrogen oxides and the oxidation of SO_2 to SO_3. If reaction kinetics phenomena are to be the subject of research, then it appears necessary that the model closely simulate prototype profiles of temperature and residence time. In such a model the velocities and turbulence will be lower than in prototype units. However, prototype units are usually designed for higher turbulence

than required for satisfactory combustion so as to minimize furnace size. It is likely that the reduced turbulence of the model will still be adequate to provide satisfactory combustion.

Basically, two scaling criteria are available for consideration in planning the design of the combustion test rig to model full-scale units. These two criteria are:

(1) constant combustion intensity
(2) constant inlet pressure drop or air velocity.

The first of these criteria provides a model having residence times similar to those in the full-scale unit. Turbulence in this model is less than in the full-scale unit. The second criteria provides a model having turbulence comparable to the full-scale unit. However, this high turbulence for the model results in combustion being completed rapdily in a small volume so that residence times are much shorter than in full-scale units.

Illustration of each criteria by the use of an example will aid in evaluating the merits of each model. Consider a full-scale axial flow furnace of firing rate F', length L', square in cross-section having width W', and M burners per row. This furnace can be divided in M^2 sub-furnaces with two types of walls, conducting and perfectly insulated. Each sub-furnace would have a firing rate F, where $F = \frac{F'}{M^2}$; length L, where $L = L'$; and width and height W, where $W = \frac{W'}{M}$. This sub-furnace will have 0 to 4 conducting walls and the remainder perfectly insulated walls.

For example in Figure 19, consider a furnace of dimensions of 14.4 ft x 14.4 ft x 22.4 ft (W' x W' x L') with four burners (M^2) and firing at a rate of 140,000,000 Btu/hr (F') or 940 gph oil. The combustion intensity is 30,000 Btu/ft^3-hr.

Now for the sub-furnace: $F = \frac{F'}{M^2} = 235$ gph oil

$$W = \frac{W'}{M} = \frac{14.4}{2} = 7.2 \text{ ft}$$

$$L = L' = 22.4 \text{ ft.}$$

The flow pattern in the furnace might be considered to consist of a highly turbulent recirculation region followed by a plug flow region. It will also be considered that the flow in each sub-furnace is identical and independent and also consists of a turbulent recirculation zone and a plug flow zone. The modeling problem can now be simplified to modeling the one burner sub-furnace rather than the multiple-burner unit. To avoid confusion of the model or test rig with the sub-furnace in the following discussions of modeling and combustion test rig design, the sub-furnace will henceforth be spoken of as the "full-scale unit" or "prototype".

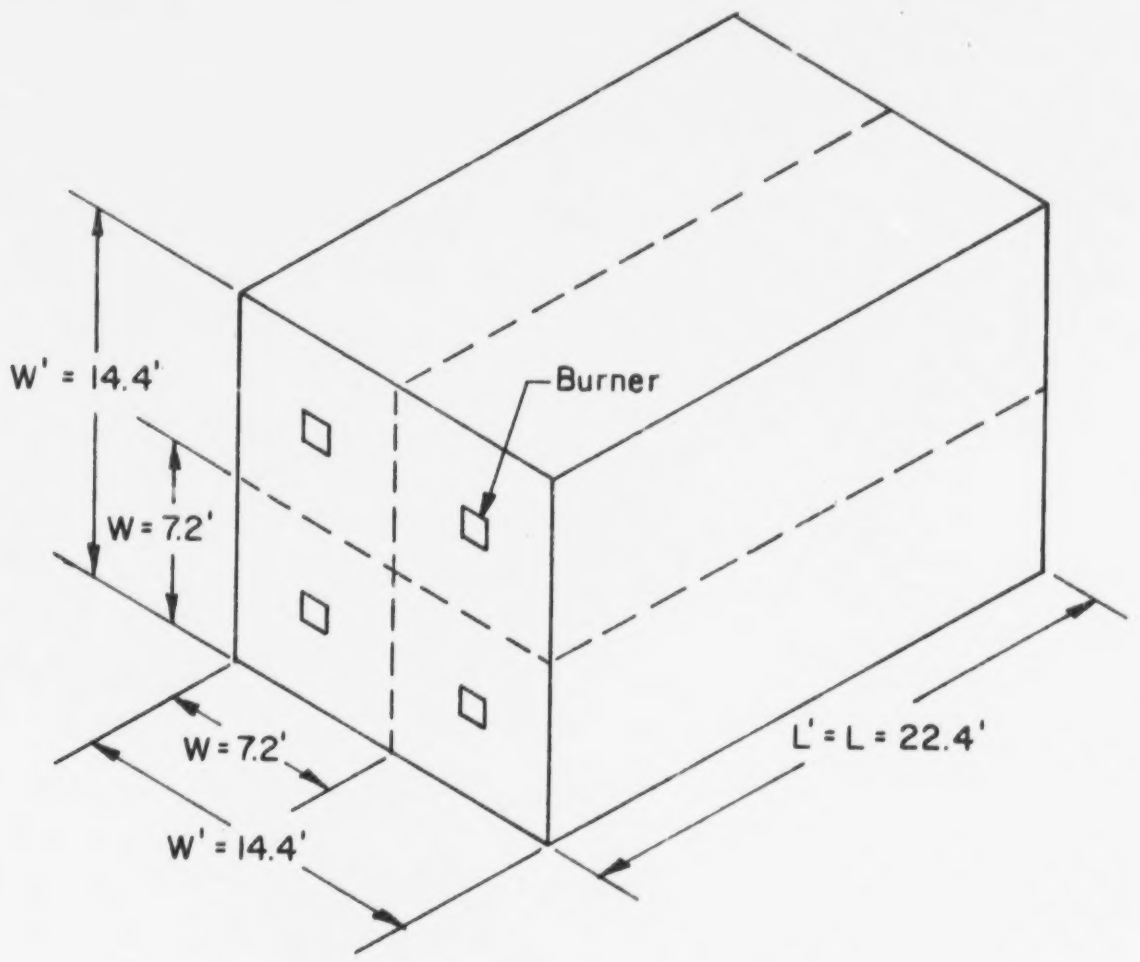


FIGURE 19. FULL-SCALE FURNACE (4 BURNERS) DIVIDED INTO 4 SUB-FURNACES FOR MODELING PURPOSES

Consideration can now be given to the two scaling criteria mentioned previously:

(1) constant combustion intensity

(2) constant pressure drop.

Constant Combustion Intensity Model.

A constant combustion intensity model would be designed for a volume firing rate of 30,000 Btu/ft^3-hr, identical with the combustion intensity of the full-scale unit.

To preserve the flow pattern of the furnace, the geometric proportions of the model and the full-scale unit should be identical. To maintain constant geometric proportions, each dimension of the sub-furnace should be reduced by a scale factor n. Therefore, the volume of the model will be related to the volume of the full-scale unit (or sub-furnace) by n^3. Maintaining combustion intensity, the ratio of fuel flow rates will be proportional to volume ratio and thus to n^3. The minimum firing rate for which satisfactory nozzle performance can be achieved is about 1.5 gph, so this rate is chosen for the model.

Therefore, if $n = \frac{W}{w} = \frac{L}{\ell}$

$$n^3 = \frac{W^2 L}{w^2 \ell} = \frac{235}{1.5} = 157$$

$$n = (157)^{1/3} = 5.4$$

then

$$w = \frac{W}{n} = \frac{7.2 \text{ ft}}{5.4} = 1.33 \text{ ft} = 16 \text{ inches}$$

$$\ell = \frac{L}{n} = \frac{22.4 \text{ ft}}{5.4} = 4.17 \text{ ft} = 50 \text{ inches.}$$

Gas velocity (U for the full-scale unit and u for the model) is proportional to volumetric throughput of gas (Q for the full-scale unit and q for the model) and inversely proportional to cross-section area. Therefore, for the full-scale unit (K is a constant):

$$U = \frac{KQ}{W^2}$$

and for the model

$$u = \frac{Kq}{w^2} .$$

Additionally, the volumetric throughput of gas is proportional to the firing rate. Because the combustion intensity is held constant, the volumetric throughput of gas is also proportional to furnace volume (V for the full-scale unit and v for the model). Then,

$$\frac{q}{v} = \frac{Q}{V}$$

and

$$q = Q\frac{v}{V} = Q\frac{v}{vn^3} = \frac{Q}{n^3}$$

It follows that,

$$u = \frac{Kq}{w^2} = \frac{q}{w^2}\frac{UW^2}{Q} = U\frac{q}{(qn^3)} \cdot \frac{(w^2n^2)}{w^2} = \frac{u}{n} = \frac{U}{5.4} .$$

Combustion intensities (C_{fs} for the full-scale unit and C_m for the model) were set equal. Therefore,

$$C_m = C_{fs}$$

where

$$C_m = H\frac{q}{v}$$

and

$$C_{fs} = H\frac{Q}{V}$$

and H is a constant.

Reynolds numbers (R_{fs} for the full-scale unit and R_m for the model) are related by

$$R_{fs} = P'WU$$

$$R_m = P'wu$$

where P' is a fluid property factor.

So that

$$R_{fs} = \frac{P'WKC_{fs}L}{H} = WC_{fs}LK'$$

and

$$R_m = \frac{P'WKC_{m}\ell}{H} = wC_{m}\ell K' = C_m K' \frac{WL}{n^2}$$

$$= \frac{R_{fs}}{n^2} = \frac{R_{fs}}{29}$$

The Reynolds number for the model is only 1/29 that of the full-scale unit.

Considering residence time (T for the full-scale unit and t for the model):

$$T = \frac{L}{U}$$

and

$$t = \frac{\ell}{u}$$

but

$$t = \frac{\frac{L}{n}}{\frac{U}{n}} = \frac{L}{U} = T \ .$$

Therefore, residence times are equal in both furnaces.

A final useful factor in evaluating modeling methods is the location of the flame with respect to the furnace length. Flame location will be related quite closely to other variables such as the location at which the air jet fills the furnace. It may be assumed that the angle of expansion of the air jet (assuming a point source) is independent of Reynolds number. Although it might appear at first glance that the air jet expansion angle would also be dependent on axial velocity, this is not what occurs. The assumption of air-jet expansion angle being independent of Reynolds number implies that the increased axial velocity (which would tend to reduce the expansion angle) is offset by increased turbulence intensity present with the higher velocities.

Hence, the distance from inlet to the location at which the jet fills the furnace (X for the full-scale unit and x for the model) is directly proportional to width and height of the furnace. Therefore, for the full-scale

unit:

$$X = BW \text{ and } \frac{X}{L} = \frac{BW}{L} \text{ (B is a constant).}$$

For the model:

$$x = Bw \text{ and } \frac{x}{\ell} = \frac{Bw}{\ell} .$$

Continuing,

$$\frac{x}{\ell} = \frac{B\left(\frac{W}{n}\right)}{\left(\frac{L}{n}\right)} = \frac{BW}{L} = \frac{X}{L} .$$

Therefore, the flame shape is identical in the model and full-scale unit.

Constant-Pressure-Drop Model. A constant-pressure-drop model would be designed for the same air velocity through the inlet wall in both the model and the full-scale unit. Using symbols as previously defined, but realizing that numerical values for the model variables will be different than for the constant combustion intensity model, the following analysis can be made.

Constant pressure drop requires that U = u.

Volume throughput of gas is proportional to gas velocity and cross-section area,

$$Q = \frac{UW^2}{K}$$

and

$$q = \frac{uw^2}{K} = \frac{U}{K}\left(\frac{W}{n}\right)^2 = \frac{1}{n^2} \cdot \frac{UW^2}{K} = \frac{Q}{n^2}$$

and

$$n^2 = \frac{Q}{q}$$

but

$$\frac{Q}{q} = \frac{235}{1.5} = 157$$

therefore

$$n^2 = 157$$

and

$$n = \sqrt{157} = 12.5$$

consequently

$$w = \frac{W}{n} = \frac{7.2 \text{ ft}}{12.5} = 0.58 \text{ ft} = 6.9 \text{ inches.}$$

To maintain identical flow patterns in the model and full-scale unit, dimensions must be proportional. Therefore,

$$\ell = \frac{L}{n} = \frac{22.4 \text{ ft}}{12.5} = 1.8 \text{ ft} = 21.5 \text{ inches.}$$

Combustion intensity for the full-scale unit is

$$C_{fs} = H\frac{Q}{V} = H\frac{\left(\frac{UW^2}{K}\right)}{(W^2L)} = \frac{HU}{KL}$$

and for the model

$$C_m = H\frac{q}{v} = H\frac{\left(\frac{uw^2}{K}\right)}{(w^2\ell)} = \frac{Hu}{K\ell} = \frac{HU}{K(\frac{L}{n})} = n\frac{HU}{KL} = nC_{fs} = 12.5\ C_{fs}\ .$$

Therefore, the combustion intensity for the model is 12.5 times the combustion intensity of the full-scale unit.

Reynolds numbers are related by

$$R_{fs} = P'WU$$

and

$$R_m = P'wu = P'\left(\frac{W}{n}\right)U = \frac{P'WU}{n} = \frac{R_{fs}}{n} = \frac{R_{fs}}{12.5}\ .$$

Therefore, the Reynolds number of the model is roughly one-twelfth that of the full-scale unit.

Finally, considering residence times

$$T = \frac{L}{U}$$

and

$$t = \frac{\ell}{u} = \left(\frac{L}{n}\right)\cdot\frac{1}{U} = \frac{1}{n}\cdot\frac{L}{U} = \frac{1}{n}T = \frac{1}{12.5}\ T\ .$$

Therefore, the residence time in this model is about one-twelfth of that of the full-scale unit.

Considering flame shape

$$\frac{x}{\ell} = \frac{Bw}{\ell} = \frac{B\,\frac{W}{n}}{\frac{L}{n}} \quad \frac{BW}{L} = \frac{X}{L}$$

and the flame shape is accurately modeled.

Quasi-Constant-Combustion-Intensity Model

A third model (referred to as the quasi-constant-combustion-intensity model) can be constructed by using the same velocity and cross-section area as the constant pressure drop model but increasing the length ℓ to equal L. Then

$$C_m = H\frac{q}{v} = H\cdot\frac{\frac{uw^2}{K}}{w^2 L} = \frac{HU}{KL} = C_{fs}$$

and $$t = \frac{L}{u} = \frac{L}{U} = T$$

with apparent combustion intensity and residence time in the model and full-scale units being equal.

However, for this model

$$\frac{x}{\ell} = \frac{Bw}{\ell} = \frac{B\cdot\frac{W}{n}}{L} = \frac{1}{n}\cdot\frac{BW}{L} = \frac{1}{n}\frac{X}{L} = \frac{1}{12.5}\frac{X}{L} \quad .$$

The flame is quite distorted in the model and, in fact, combustion is limited to the space near the inlet wall. Added length, which appears to reduce combustion intensity, in reality only serves as a post-combustion space.

Comparison of Models With the Full-Scale Unit

The following table summarizes the relationship of important variables for the modeling criteria discussed above. It is obvious that each model has its advantages and shortcomings.

The constant combustion intensity model provides the desired residence times and flame shape, but at a lower turbulence level (Reynolds number).

	Full-Scale Unit	Constant Combustion Intensity	Constant Pressure Drop	Quasi-Constant Intensity Model
Width = Height	W	$\frac{W}{5.4}$	$\frac{W}{12.5}$	$\frac{W}{12.5}$
Length	L	$\frac{L}{5.4}$	$\frac{L}{12.5}$	L
Velocity	U	$\frac{U}{5.4}$	U	U
Residence Time	T	T	$\frac{T}{12.5}$	T
Combustion Intensity	C_{fs}	C_{fs}	$12.5\ C_{fs}$	C_{fs}
Reynolds Number	R_{fs}	$\frac{R_{fs}}{29}$	$\frac{R_{fs}}{12.5}$	$\frac{R_{fs}}{12.5}$
Flame Shape	$\frac{X}{L}$	$\frac{X}{L}$	$\frac{X}{L}$	$\frac{1}{12.5}\ \frac{X}{L}$

Although the turbulence level is reduced for this model, most practical furnaces are designed for turbulence levels far in excess of the turbulence necessary for combustion. This is done to reduce flame length and allow construction of small, space-efficient units. The turbulence level designed for our model should be adequate for the purposes of this study.

The constant-pressure-drop model maintains flame shape but both residence time and turbulence level are reduced. An order of magnitude reduction in residence time would severely affect the validity of pollutant reaction data.

Finally, the quasi-constant combustion intensity model apparently preserves the residence time of the full-scale unit but at a loss in turbulence and with a distorted flame shape. Although the residence time is apparently preserved, the change of flame shape results in most of this residence time

being in the post-combustion region and not the combustion region as desired. The combustion region residence time would be essentially identical to that of the constant-pressure-drop model.

Brown and Thring[36] considered two types of modeling in connection with the use of pressure-jet burners in marine boilers. They list the variation of several parameters with scale for the two cases of constant velocity (constant pressure drop) and constant residence time (constant combustion intensity). Agreement between parameters for their hot model and for the full-scale unit appears slightly better for the constant-residence-time model, but agreement is satisfactory in both cases.

It is possible to provide a reasonable simulation of the constant combustion intensity and the constant pressure drop models in a single model combustion chamber by using two levels of firing rate. A low rate, such as 1.5 gph, would provide full-scale unit residence times, while a higher firing rate, such as 8 gph, would simulate velocities and mixing effects comparable to the full-scale units. Because of the relatively large size of the combustion space for the residence-time simulation, the heat loss to the chamber walls will be a critical factor in simulating flame temperature. In order to avoid excessive chilling of the flame by wall radiation it will be necessary to provide effective insulation or, perhaps, auxiliary heating in the walls to limit heat loss to about 4500 Btu/ft^2-hr. However, for proper temperature conditions when firing at the higher rate simulating constant-velocity conditions, wall heat loss should be about 22,000 Btu/ft^2-hr.

The foregoing discussion is based upon the premise that velocities in the combustion space are determined by size of the space. Actually, they are controlled by burner design, and it is possible to use a turbulent burner of small size to form a small flame in a large combustion space. Accordingly, it appears quite feasible to simulate both residence time and turbulence levels in the same furnace model at about the same firing rates by using two different burners having widely different velocities. A large, low-velocity burner would

form a large flame that fills the combustion space, and a small, high-velocity burner would form a small, highly-turbulent flame that occupies about 1/5 of the available furnace volume. Using this expedient, it is probable that either residence time or velocity can be simulated at about the same firing rate in the same model furnace.

Other Scaling Phenomena

Several other scaling phenomena (nozzle scaling, heat flux, and buoyancy) are discussed in the Appendix. Summarizing these discussions:

1. Nozzle scaling: To obtain the same droplet size as in a full-scale unit would require reducing air pressure (assuming two fluid atomizers) and/or reducing fuel viscosity for the model.

2. Heat flux: The insulating value of all furnace walls of the model will have to be increased relative to the full-scale unit.

3. Buoyancy: Buoyancy effects in the model will be greater than in the full-scale unit but should not be a problem.

Design of Combustion Test Rig

Figure 20 summarizes the modeling considerations discussed in the foregoing sections. The approach starts with the full-scale industrial boiler and moves step-by-step through a sub-section of the full-scale furnace to a square-sectioned model and, finally, to a circular equivalent model for the proposed combustion test rig.

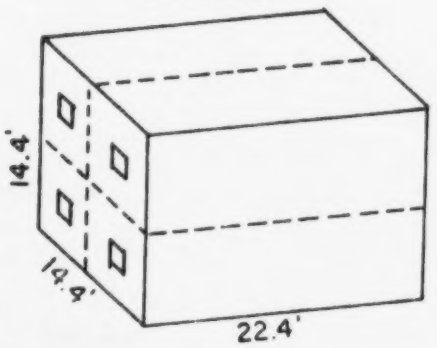
Furnace Design

Figure 21 is a section view of the furnace for the proposed combustion test rig. It consists of a cylindrical cavity 18 inches in diameter and 50 inches long. Dimensionally, this furnace is similar to the furnace with 16 inch x 16 inch cross section. However, the cylindrical shape provides a more symmetrical model which eliminates corner effects. The outer shell of the rig is 10-gage steel and is airtight; the furnace is lined with insulating firebrick 4.5 inches thick, with block-type insulation between the brick and the shell. Several sampling ports in one side will provide for traversing across the flame at various distances from the burners. An axial port, at the outlet, will permit axial traversing should this prove desirable.

The outlet diameter for the model furnace would be reduced to six inches to enclose the flame and provide for normal recirculation and normal radiation losses.

The space within this furnace is designed for a combustion intensity of 30,000 Btu/ft^3-hr when firing at a rate of 222,000 Btu/hr or about 1.5 gph of No. 6 fuel oil. Residence time in the furnace will be about 1.7 sec, comparable to residence times in the full-scale units mentioned earlier. For time-temperature relations in the model comparable to that of the prototype, the heat-transfer rate to the walls should be about 4500 Btu/ft^2-hr when firing to simulate residence time, and about 22,000 Btu/ft^2-hr when firing to simulate furnace velocity.

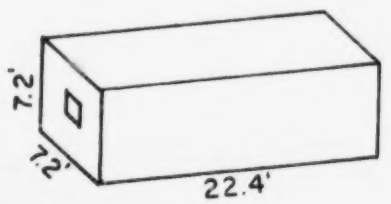
Furnace



Full-scale industrial furnace

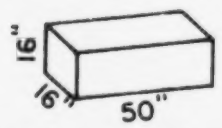
Operational Parameters

Firing rate: 140,000,000 Btu/hr
Combustion intensity: 30,000 Btu/ft^3-hr
Firing rate / Cross section: 675,000 Btu/ft^2-hr



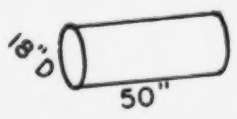
Sub-furnace

Firing rate: 35,000,000 Btu/hr
Combustion intensity: 30,000 Btu/ft^3-hr
Firing rate / Cross section: 675,000 Btu/ft^2-hr



Constant combustion intensity model

	Constant Combustion Intensity	Constant Pressure Drop
Firing rate:	222,000 Btu/hr	1,200,000 Btu/hr
Combustion intensity:	30,000 Btu/ft^3-hr	162,000 Btu/ft^3-hr
Firing rate / Cross section:	125,000 Btu/ft^2-hr	675,000 Btu/ft^2-hr



Cylindrical equivalent of constant combustion intensity model

Firing rate:	222,000 Btu/hr	1,200,000 Btu/hr
Combustion intensity	30,000 Btu/ft^3-hr	162,000 Btu/ft^3-hr
Firing rate / Cross section:	125,000 Btu/ft^2-hr	675,000 Btu/ft^2-hr

FIGURE 20. MODELING PROCEDURE TABULATING OPERATIONAL PARAMETERS FOR FULL-SCALE FURNACES AND MODELS

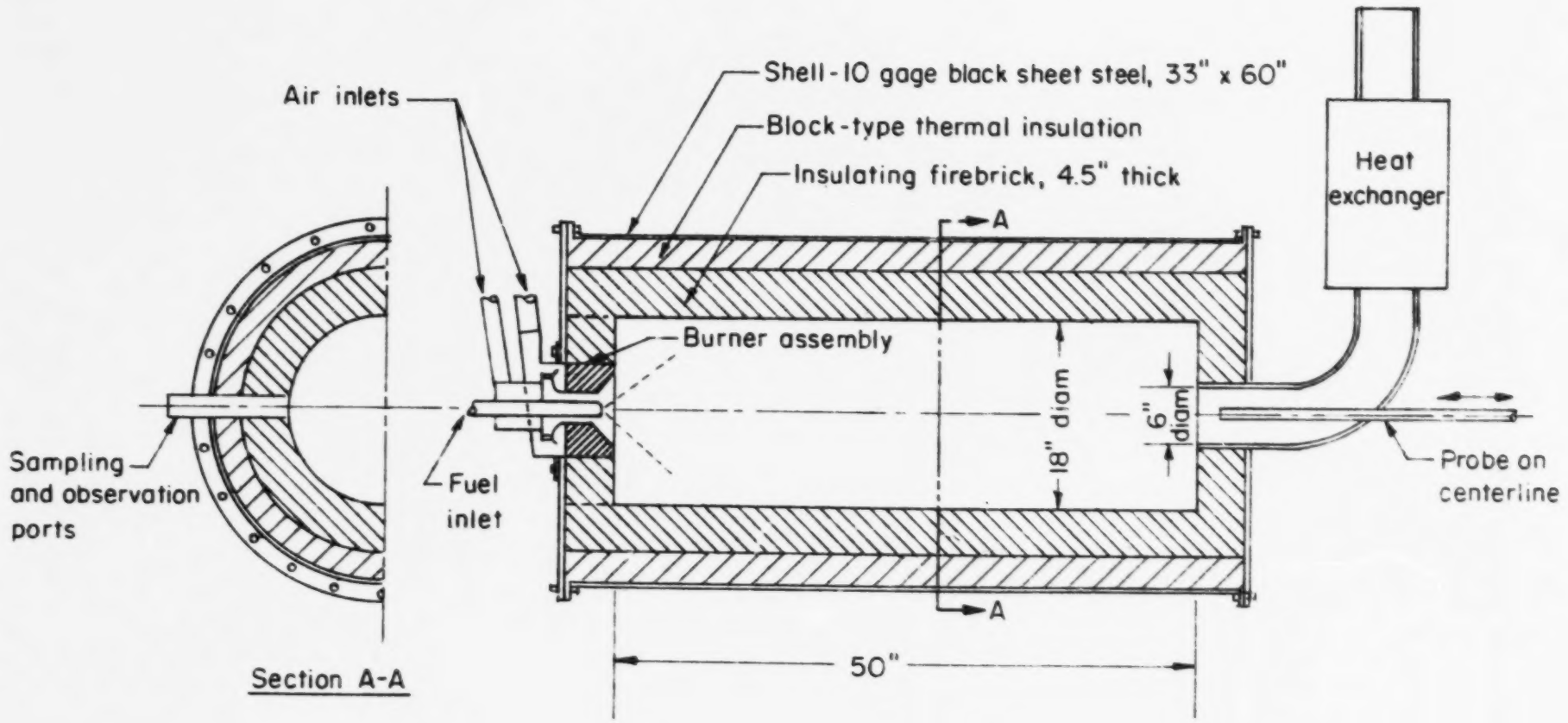


FIGURE 21. FURNACE SECTION OF COMBUSTION RIG

To simulate time-temperature gradients to lower temperatures, as for the passes through the boiler heat-transfer surfaces, the furnace length would be extended and/or cooling surface added to obtain gas cooling at the desired rate.

Burner Design

Figure 22 is a cross-section of the model burner. The burner incorporates considerable flexibility to permit operation over a range of conditions. It is possible to vary the firing rate, the flame velocity, the swirl angle, and the type of fuel fired. When all air is admitted through the axial holes in the plate to the left of the burner throat in Figure 22, all flow will be axial. The turbulence level in this axial stream will be relatively high because air is admitted at a velocity of about 50 fps or more through 24 holes; the jets from these holes will merge in the burner throat to provide a highly turbulent stream at low axial velocity. By adjustment of the proportion of air flow to the swirl vanes and to the axial holes, the percentage of swirl in the burner throat can be adjusted at will. Because of the high turbulence from the axial-flow holes, the overall turbulence level will remain about the same as the tangential velocity is reduced.

The burner throat of 3.5-inches diameter is designed for an axial velocity of 15 fps, to produce a large flame that will fill the furnace. In order to simulate velocities in a full-scale boiler burner, two alternatives are available. First, the firing rate can be increased by a factor of five (to 8 gph) with no other changes. The flame size and shape will be about the same as with the low firing rate, but the residence time will be reduced by a factor of five. Alternatively, an insert can be placed in the burner throat (see Figure 22), reducing its diameter from 3.5 inches to 2.875 inches. (This insert can be a metal ring held in place by set screws.) The throat velocity, with this change, will increase by the required factor of about 5 so that full prototype velocity will be obtained at the 1.6 gph firing rate. With this change, the flame should become much smaller and will not fill the combustion space.

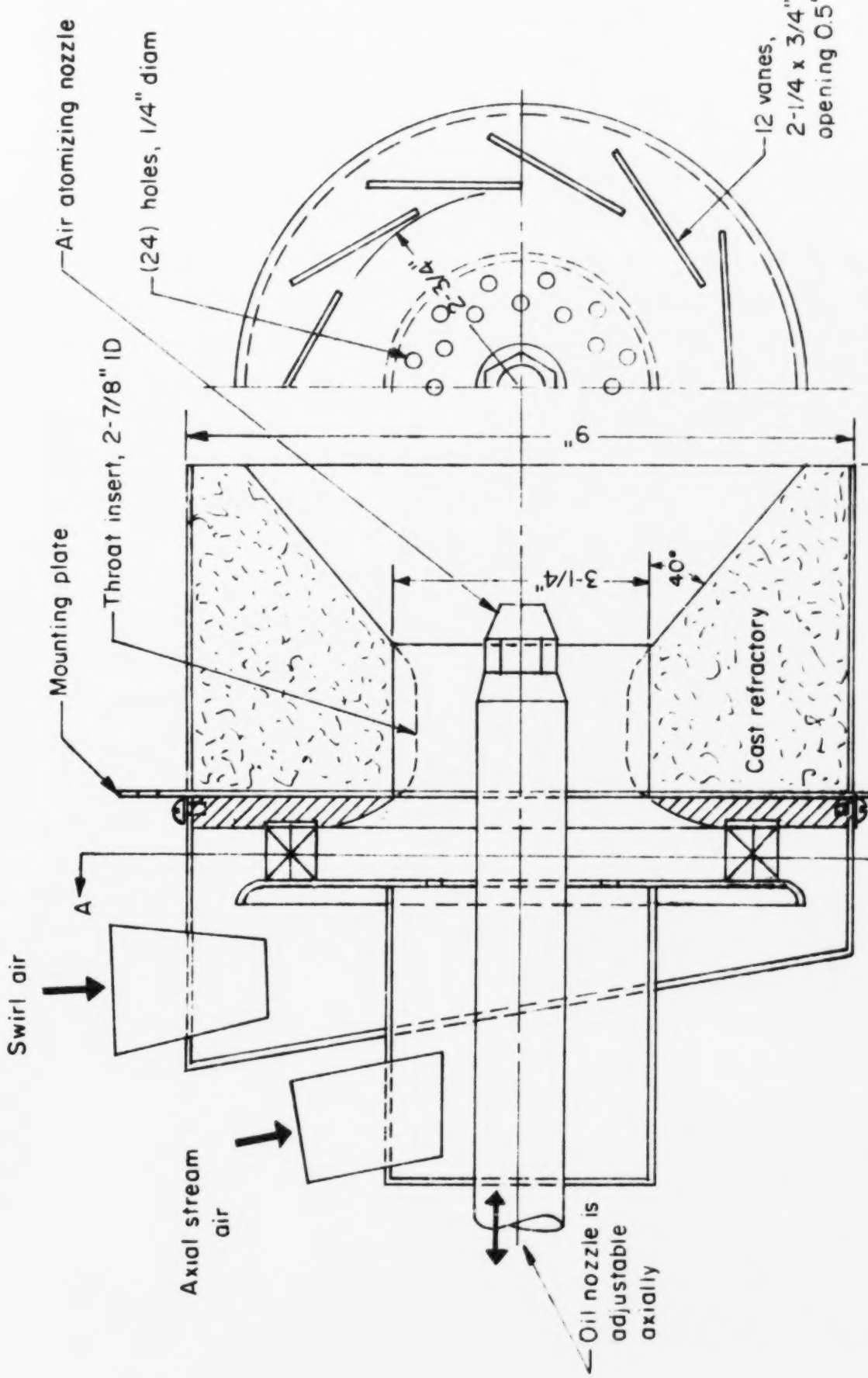


FIGURE 22. DETAIL OF BURNER FOR COMBUSTION RIG

The fuel nozzle shown is a Delavan "Industrial Aero" air-atomizing nozzle which was used for the "cold" atomization trials of No. 6 emulsions reported earlier.

A combustion rig of the type discussed above is sufficiently versatile to have a wide variety of uses. Other possible uses of this rig include studies of the effect of combustion variables such as droplet size, air-fuel ratio, and external recirculation of flue gas on pollutant emissions. With the addition of controlled-temperature heat-transfer tubes, the rig would also be useful for studying the effects of additives on superheater tube corrosion and deposits.

It is possible to fire natural gas in this burner by replacing the fuel nozzle with a gas injector. Pulverized coal can also be fired by substituting a coal nozzle for the oil atomizer. The coal nozzle would require a central cone to form a conical coal dispersion much like an oil spray. In a previous furnace of this size at Battelle, coal was fired at rates from 50 to 100 lb/hr.

Auxiliary Equipment

Figure 23 shows a schematic layout of the laboratory setup required to support the rig. Included are a source of burner air, two parallel air flowmeters and control valves for the two burner air inlets,and a fuel pump and preheater to heat the oil to the required atomizing temperature of about 250 F. Instrumentation is also required for sampling and analysis of furnace and exhaust gas and for temperature measurement. Fuel rate would be controlled by operation of a variable-displacement pump and measured by weighing.

Estimated Construction Cost

The estimated cost for constructing and checking out an oil-fired combustion test rig as described above and placing it in operation is approximately $48,000. This estimate is based on costs at Battelle-Columbus including shop, technician, technical supervision and overhead.

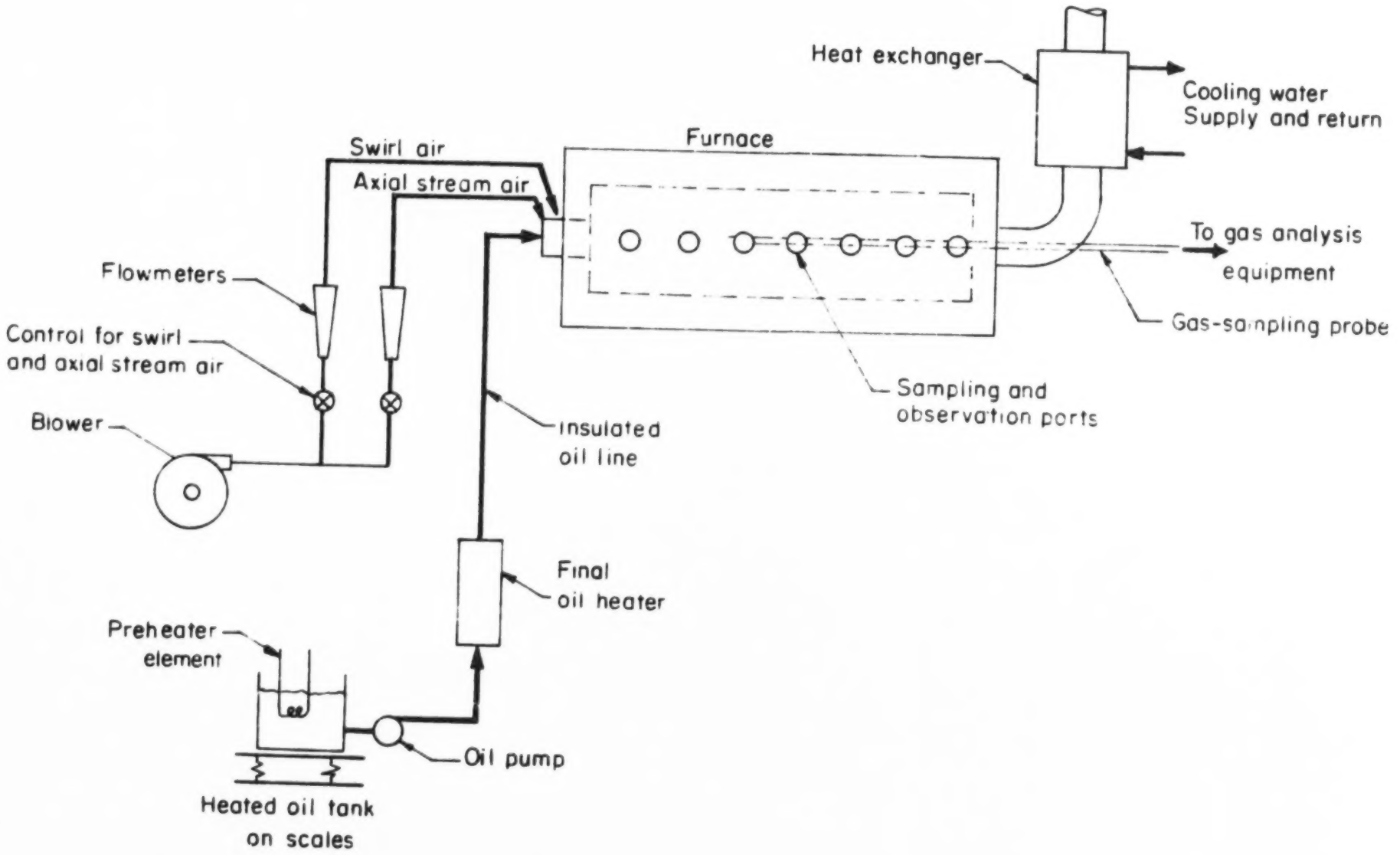


FIGURE 23. LABORATORY SETUP FOR COMBUSTION RIG

It assumes that the burner design has been established through preliminary trials and is not included in the cost estimate. Costs associated with gas analysis and emission measurements are not included.

The total cost estimate consists of the following items:

1. Final detailed design of the burner, furnace, heat exchanger, and instrumentation
2. Construction of burner
3. Construction of furnace
4. Construction of heat exchanger
5. Assemble rig with oil preheat system, controls and instrumentation
6. Sufficient firing and checkout to select operational parameters at various firing rates.

ACKNOWLEDGMENTS

Contributions of the following Battelle staff members are acknowledged: Dr. David B. Cox in determining viscosity and rheological properties, Robert D. Giammar and James J. Tavor in performing the "cold" atomization trials, Dr. James A. Gieseke in examining atomized droplets, and William T. Reid in evaluating the corrosion and deposits tendencies.

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APPENDIX

APPENDIX

A. INCORPORATION OF BASIC ADDITIVES IN WATER-IN-DISTILLATE FUEL OIL EMULSIONS

One aspect of this task order was to determine if basic additives could be incorporated into water-in-distillate fuel oil emulsions such as prepared under the previous task order[3].

In the earlier work, a surfactant which promoted stable water-in-distillate oil emulsion was identified. The surfactant consisted of 4 parts (by weight) of sorbitan sesquioleate and 1 part polyoxyethylene (20) sorbitan monopalmitate.

In this task, relatively stable water-in-distillate oil emulsions containing 20 weight percent water and 10 weight percent NH_4OH, $Ca(OH)_2$ or $Mg(OH)_2$ were prepared using this surfactant and the procedures described in the report on Task Order No. 8[3]. However, the surfactant is not compatible with NaOH. A suitable emulsion containing NaOH was prepared using about 5 weight percent naphthenic acid as an emulsifying agent. (Naphthenic acids are constituents of crude and distillate oils.)

None of these water-in-distillate fuel oil emulsions containing basic additives exhibited long-term stability. However, they appeared to be suitable for preliminary burner tests.

B. OTHER SCALING CONSIDERATIONS

Three other factors which must be considered in modeling and choice of conditions are (1) nozzle scaling, (2) wall heat flux and (3) buoyancy of hot gases. These additional considerations are discussed in this Appendix.

Nozzle Scaling

In choosing a two-fluid nozzle for t'.e combustion rig, the question occurs as to what is the relationship between the atomization in the full-scale boiler and that in the combustion rig. It is assumed that the flow rate of atomizing air for the model nozzle and the full-scale or prototype nozzle do not have to be in the same ratio as the firing rates, as each is only a small fraction of the total air flow in the corresponding combustion chamber.

Momentum Flux. To obtain the same momentum flux ratio (atomizing air to total combustion air) in the model and the prototype, and assuming that the fuel adds no momentum when it is injected or aspirated by the air flow in the nozzle,

$$\frac{\left(\frac{\rho_a q_a^2}{d^2}\right)}{\left(\frac{\rho_c q^2}{w^2}\right)} = \frac{\left(\frac{\rho_a Q_a^2}{D^2}\right)}{\left(\frac{\rho_c Q^2}{W^2}\right)}$$

where d = characteristic nozzle dimension for a series of geometrically similar nozzles (d for model nozzle, D for full-scale unit nozzle)

q_a and Q_a = volumetric air flow rates through nozzles of the model and prototype

ρ_a = density of atomizing air

ρ_c = density of combustion air.

For the constant-combustion-intensity model, this reduces to,

$$n^2 = \left(\frac{Q_a}{q_a}\right)\left(\frac{d}{D}\right).$$

As n is fixed, it is seen that this equation expresses a relationship between the atomizing air-flow rate and the characteristic dimension of the nozzle. The exact range of dimensions and flow rates to be considered would result from comparison with various prototype designs.

Further, the nozzle for the model is expected to be fired at only 1/m of the design maximum firing rate of the nozzle. Thus, $q = \frac{1}{m} q_d$. If the design firing rate for a series of nozzles is proportional to d^2, and assuming the prototype is fired at the design firing rate for the nozzle, then

$$\frac{D^2}{d^2} = \frac{Q}{q_d}$$

and

$$D^2 = d^2 \left(\frac{Q}{q_d} \right) = d^2 \frac{Q}{m \frac{Q}{n^3}} = d^2 \left(\frac{n^3}{m} \right) .$$

Therefore,

$$\frac{Q_a}{q_a} = n^{7/2}/m^{1/2} .$$

However, by direct scaling of the nozzle dimensions and for constant atomizing air pressure,

$$\frac{Q_a}{q_{a,d}} = n^2$$

and thus,

$$\frac{q_{a,d}}{q_a} = n^{3/2}/m^{1/2} .$$

Therefore, the atomizing air pressure for nozzle of the combustion rig would have to be reduced by a factor

$$(n^{3/2}/m^{1/2})^{1/2} \text{ or } \left(\frac{n^3}{m} \right)^{1/4} .$$

Obviously, if the nozzle for the model is fired at its design maximum firing rate, then m reduces to 1.0 in all above equations.

Droplet Size. From Wigg's study of drop size from several two-fluid atomizing nozzles[37], it is found that drop size is a function of

$$(\gamma^{.5} \sigma^{.2}) \; (\rho_f q_f)^{.1} \; \frac{(\rho_a q_a + \rho_f q_f)^{.5} D^{.1}}{\rho_a q_a} / U_i$$

where

γ_m and γ_p = viscosity of oil in model and prototype

σ_m and σ_p = surface tension of oil in model and prototype

ρ_f = density of oil

ρ_a = density of atomizing air

q_f and Q_f = volumetric flow rate of oil in model and prototype

U_i = characteristic nozzle internal air velocity.

To obtain constant drop size and constant burning time for the model and the prototype, this relationship must be constant. Thus, after replacing U_i by $\frac{Q_a}{D^2}$,

$$\left[\left(\frac{\gamma_m}{\gamma_p}\right)^{.5}\left(\frac{\sigma_m}{\sigma_p}\right)^{.2}\right]\left[1+\frac{\rho_f q_f}{\rho_a q_a}\right]^{1/2}\left[\frac{\rho_f q_f}{\rho_a q_a}\right] n^{2.7} =$$

$$\left[1+\frac{\rho_f Q_f}{\rho_a Q_a}\right]^{1/2}\left(\frac{D}{d}\right)^{2.1}\left(\frac{\rho_f Q_f}{\rho_a Q_a}\right)$$

Finally, if the relationship between size and air flow rate in the nozzle developed earlier is used,

$$\left[\left(\frac{\gamma_m}{\gamma_p}\right)^{.5}\left(\frac{\sigma_m}{\sigma_p}\right)^{.2}\right]\left[1+\frac{\rho_f q_f}{\rho_a q_a}\right]^{1/2} n^{1.7} =$$

$$\left[1+\frac{\rho_f Q_f}{\rho_a Q_a}\right]^{1/2}\left(\frac{D}{d}\right)^{1.1} .$$

It may not be possible to satisfy this equation (which provides constant droplet size) and the momentum flux relationship simultaneously while also holding γ and σ constant. This point would have to be checked with more detailed nozzle analysis including examination of operational parameter relationships for prototype nozzles.

Heat Flux

The heat flux to the walls should also be considered. For the constant combustion intensity model, the firing rate, and thus heat input, scales according to n^3. The heat loss varies with area. Thus, to maintain the ratio of heat loss to heat input,

$$\frac{\left(\frac{w\ell(\Delta T_m)\, k_m}{y}\right)}{C_m w^2 \ell} = \frac{\left(\frac{WL(\Delta T_p)\, k_{fs}}{Y}\right)}{C_{fs} W^2 L}$$

where y and Y are the wall thickness is of the model and full-scale unit and ΔT_m and ΔT_{fs} are temperature drops through the furnace wall for the model and full-scale unit. If $\Delta T_m = \Delta T_{fs}$, this leads to

$$n = \frac{y}{Y} \cdot \frac{k_{fs}}{k_m} \quad .$$

With typical values, a magnesia wall in the model could replace a steel wall of the same thickness in the full-scale unit. Considering the case of an insulating wall in the prototype, the problem becomes more severe. First, it is noted that the thickness, y, now becomes a matter of inches rather than fractions of an inch. Furthermore, easily available materials with much lower k values are not available.

However, if $\Delta T_m < \Delta T_{fs}$, then

$$n = \frac{y}{Y} \cdot \frac{k_{fs}}{k_m} \cdot \frac{\Delta T_{fs}}{\Delta T_m} \quad .$$

Hence, reasonable values of y and k_m could be obtained by decreasing ΔT_m considerably. This could be accomplished by heating the outside walls of the model.

There is another aspect to the heat conduction that must be considered. That is the transfer of heat to the wall from the gas. Roughly speaking, if the gas temperature profile is the same, the ratio of heat transfer to the walls to the heat input varies with Reynolds number to the minus 0.2 power. Thus, the loss on this basis would be 0.4 or about 2.0 times as much for the model as for the full-scale unit. Assuming the insulation for the model has been selected

to maintain the ratio of heat loss to heat input, the higher heat flux on the inner wall will force the inner surface temperature of the wall up in the model and partly correct the problem. However, to compensate as nearly as possible, the insulating value of the wall should be improved, by relaxing the requirement on the matching of wall surface temperature on the inside. For the final design, sample computations could be made using reasonable values for both metal and ceramic full-scale unit walls.

Buoyancy Effects

The Froude number (Fr_{fs} for the full-scale unit and Fr_m for the model) is the pertinent expression for considering the effect of buoyancy on the flow pattern. It is noted that

$$Fr_{fs} = \frac{U^2}{gL}$$

and

$$Fr_m = \frac{u^2}{g\ell} = \frac{\left(\frac{U}{n}\right)^2}{g\left(\frac{L}{n}\right)} = \frac{1}{n}\frac{U^2}{gL} = \frac{Fr_{fs}}{n}$$

for the constant combustion intensity case. Thus, the Froude number cannot be held constant (unless g varies which is impracticable). A little consideration shows that the ratio of the buoyancy-induced deflection to horizontal distance, or fractional deflection, increases with n. Thus, a fractional deflection from buoyancy effects in the prototype of 2 percent would be about 12 percent in the model. The deflection would probably not be a problem in the model.

C. NOMENCLATURE

d - characteristic nozzle dimension, nozzle for model

g - gravitational constant

k_{fs} - thermal conductivity of wall, full-scale unit

k_m - thermal conductivity of wall, model

ℓ - length of furnace cavity, model

n - scaling factor, subfurnace to model

q - gas throughput, model

q_a - atomizing air, volumetric flow rate, model

q_a,d - design atomizing air volumetric flow rate, nozzle for model

q_d - gas throughput of model for design firing rate of model nozzle

q_f - oil volumetric flow rate, model

t - residence time, model

u - gas velocity, model

v - volume of furnace cavity,model

v_a - specific volume of combustion products, average

w - width of furnace cavity, model

x - distance for jet to fill furnace, model

y - wall thickness, model

B - constant

D - characteristic nozzle dimension, nozzle for full-scale unit

F - firing rate, subfurnace

F' - firing rate, full-scale unit

Fr_{fs} - Froude number, full-scale unit

Fr_m - Froude number, model

H - constant

K - constant

K' - constant

L - length of furnace cavity, subfurnace

NOMENCLATURE (Continued)

L' - length of furnace cavity, full-scale unit

M^2 - number of burners, full-scale unit

P - pressure

P' - fluid property factor

Q - gas throughput, full-scale unit

Q_a - atomizing air, volumetric flow rate, full-scale unit

Q_f - oil volumetric flow rate, full-scale unit

R - gas constant

T - residence time, full-scale unit

T_a - absolute temperature of flue gas, average

T_o - residence time in furnace of volume V_o

U - gas velocity, full-scale unit

U_i - characteristic nozzle internal air velocity

V - volume of furnace cavity, full-scale unit

V_o - volume of a furnace

W - width of furnace cavity, subfurnace

W' - width of furnace cavity, full-scale unit

X - distance for jet to fill furnace, full-scale unit

Y - wall thickness, full-scale unit

C_{fs} - combustion intensity, full-scale unit

C_m - combustion intensity, model

R_{fs} - Reynolds number, full-scale unit

R_m - Reynolds number, model

NOMENCLATURE (Continued)

ΔT_{fs} - temperature difference across wall, full-scale unit

ΔT_m - temperature difference across wall, mocel

γ_m - viscosity of oil, model

γ_p - viscosity of oil, prototype

ρ_a - density of atomizing air

ρ_c - density of combustion air

ρ_f - density of fuel oil

σ_m - surface tension of oil, model

σ_p - surface tension of oil, prototype